# FUEL INJECTION

SOME EXPERIMENTS

on the

HYDRAULIC CHARACTERISTICS

of a

BOSCH INJECTION SYSTEM.

C O N T E N T S.

# PART I.

# INTRODUCTION

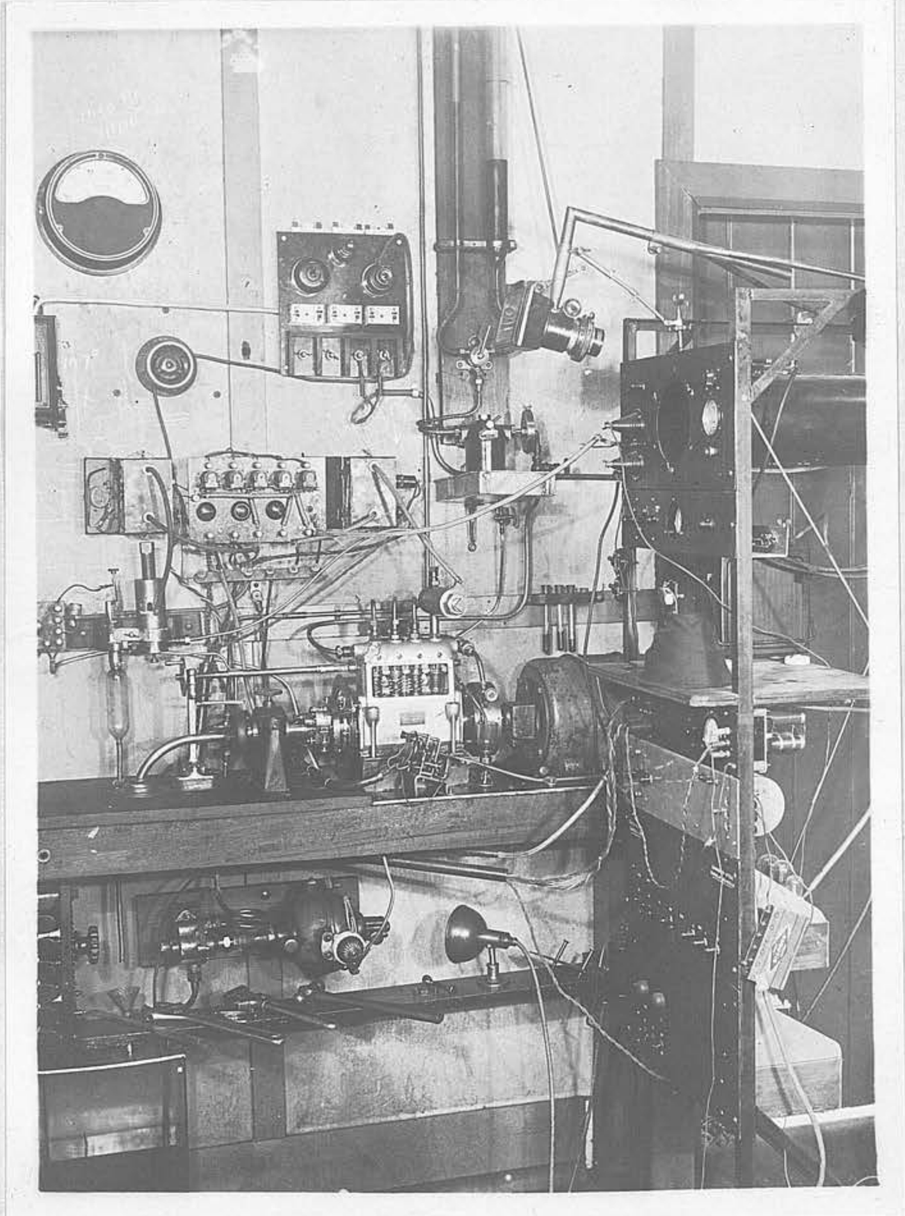

FIG. I.

GENERAL ARRANGEMENT.

## PART I. INTRODUCTION.

The present century has already witnessed a revolution as far reaching in its effects as that brought about by the development of the steam engine. Oil and the Internal Combustion Engine replacing or supplementing coal and steam have provided power so cheap and clean and convenient and made travel so easy and speedy that the isolation and inconvenience of life in the country have been largely removed, and rapid transportation of men and materials by land or sea or air become matters of everyday routine.

The development of the internal combustion engine responsible for these revolutionary changes has taken place along two well defined lines. On the one hand, the earlier type closely following established gas engine practice uses a compressed air-vapour mixture, ignition being effected by electric spark or similar device. In the more modern development, a far-reaching departure from established practice has been made. Air only is compressed in the engine cylinder and liquid fuel of a refinement dependent principally on engine speed is injected into the cylinder towards the end of compression. Combustion, effected by the heat of compression, is initiated in the neighbourhood of the more finely divided globules of fuel and rapidly spreads throughout the whole mass of the mixture.

A study of the historical and technical development of this later type of internal combustion engine contained in a paper entitled "The Origin and Development of the Heavy Oil Engine" is submitted as part of the present Thesis, and only brief reference will therefore be made to those features of development having bearing on the Research forming the subject of the thesis.

The most outstanding problems associated with oil engine development have been those dealing with the injection and combustion processes. Akroyd Stuart, who may be described as the Father of the modern oil engine failed to discover a solution; Diesel, with whose name the modern oil engine is usually associated shunned the difficulty by adopting air injection; and the multiplicity of designs employed at the present day indicate that the problems are as yet unsolved.

Injection and combustion are complementary factors in oil engine performance and unless full consideration be given to their associated effects, no study of the injection system can be looked upon as complete. Such a study calls for equipment and resources of exceptional magnitude and it would appear that the complete problem can most easily be solved by a reasoned correlation of results obtained from careful research into the separable factors.

The present research carried out partly in the University of Edinburgh and partly in the University of Western Australia under the direction of Sir Thomas Hudson-Beare, Regius

Professor of Engineering in the University of Edinburgh has been undertaken in an endeavour to throw fresh light on the injection process of the high speed oil engine using a fuel pump of the Bosch Type and a spring loaded differential needle atomiser. The work has been confined to the injection characteristics of the system, and no attempt has been made to elucidate the response of the engine to the influence of the injection process as a whole.

## PART II.

## THE NATURE of the PROBLEM.

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Direct injection has superseded all other methods of introducing the oil into the engine cylinder and high engine speeds have demanded a pump capable of injecting minute quantities of oil in extremely short time-intervals with a high degree of precision. The initial problems of pump design were chiefly those of mechanical construction, and the successful solution constitutes a triumph for modern machine shop practice. There remain, however the hydraulic problems associated with rapid impulses imparted to a fluid column, and if the injection characteristics are to be accurately determined, the significance of their effects must be gauged under the widely varying conditions of engine speed and load.

The most important factors are the compressibility, elasticity, and inertia of the fuel column and the hydraulic resistance to flow through the system. These factors separately or in combination will influence the quantity and distribution of the oil feed to the engine, and should therefore be considered in conjunction with jet dispersion, flame propagation and engine performance. But, as mentioned in the introduction, the complete analysis is a research undertaking of considerable magnitude and for the present, it seems desirable to attempt refined investigation of the separable problems. The investigation of the pressure distribution in a spring loaded nozzle injection system which forms the basis of the present research, has an intimate bearing on the quantity and form of the oil discharge and is essential to an understanding of engine performance under varying load and speed operations.

The nature of the flow from the pump to the nozzle far from being the simple process assumed by early pump designers has been shown both by theoretical considerations of the hydraulic principles involved and by experiment to be a most complicated phenomenon.

In a pump suitable for high speed engines, the nature of the pressure impulse in the fuel line is such as to involve consideration of the elasticity and inertia of the fuel as well as its compressibility. The flow of water under similar conditions has been extensively investigated and basing his calculations on the work of Allievi [1], Sass [2] has developed a mathematical theory of the pressure variation at the injection nozzle of a spring loaded jerk pump system. Under the conditions obtaining in a high speed injection system similar to that under investigation in the present work, he shows that pressure propagation in the pipe line between pump plunger and nozzle is largely affected by pressure waves originating at the plunger, reflected wholly or in part from the nozzle according to conditions of discharge obtaining there, and the process repeated alternately at pump plunger and nozzle with an intensity and duration varying for any nozzle setting with such factors as pump speed, plunger area, cam contour and pipe dimensions.

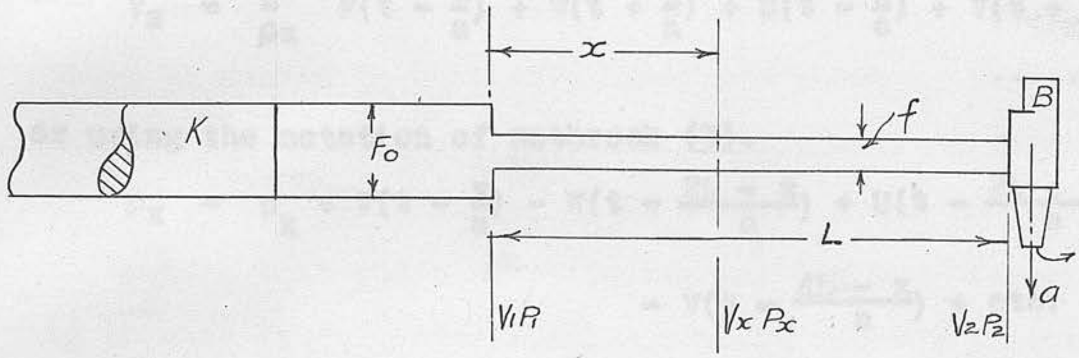



Fig. II.

In the system shown in Fig. I (reproduced from Sass' work) K is the pump plunger area Fo connected through the pipe of area f and length L to the nozzle B. The differential equations representing the condition in the system are given as:

$$\frac{\delta V}{\delta t} = -\frac{1}{\rho}\frac{\delta P}{\delta x}$$

$$\frac{\delta V}{\delta x} = -\frac{1}{\rho a^2}\frac{\delta P}{\delta t} \qquad (1)$$

where V is the velocity of the fuel at time t at point 'x' measured from the pump barrel, $\rho$ is the density of the fuel and a the velocity of pressure waves in the system. The latter is given by $a = \sqrt{\frac{Kg}{\rho}}$ where K is the bulk modulus of the fuel and $\rho$ the density.

The general integral of Equation I is given by Sass as:-

$$p_x = p_k + F(t - \frac{x}{a}) - W(t + \frac{x}{a}) + U(t - \frac{x}{a}) - V(t + \frac{x}{a}) + \text{etc.}$$

$$v_x = \frac{1}{\rho a} F(t - \frac{x}{a}) + W(t + \frac{x}{a}) + U(t - \frac{x}{a}) + V(t + \frac{x}{a}) + \text{etc.} \quad \text{.........(2)}$$

or using the notation of Rothrock (3).

$$p_x = p_k + F(t - \frac{x}{a}) - W(t - \frac{2L - x}{a}) + U(t - \frac{2L + x}{a}) - V(t - \frac{4L - x}{a}) + \text{etc.}$$

$$v_x = \frac{1}{\rho a} F(t - \frac{x}{a}) + W(t - \frac{2L - x}{a}) + U(t - \frac{2L - x}{a}) + V(t - \frac{4L - x}{a}) + \text{etc.}$$

In these latter equations, $p_x$ and $v_x$ represent the pressure and velocity respectively at a point distant x from the pump end of the pipe, L the length of the pipe and $p_k$ the initial pressure in the pipe. Pressure waves considered as originating at the pump end of the pipe and travelling with velocity "a" equal to the velocity of sound in the oil are represented by the functions F, W, U etc., and using the above notation, the pressure at any point x from the pump end of the pipe at any time t is given by the algebraic summation of the terms of the equation.

Sass[2] and Rothrock [3] have determined theoretically the significance of the functions F, W, etc. for standard injection systems, and the deductions compared with experimental results, show reasonably good agreement.

Analytical investigations of the pressure and discharge characteristics have also been made by other workers including Davies and Giffen[4], and Farmer and Alcock[5]. While the usefulness of such theoretical investigations cannot be denied in that they have been shown by experimental work to be basically correct, and therefore of value in interpreting actual performance, it is equally proved by the same experiments that there exist in any particular system of injection so many "built in" variables, that it becomes a matter of extreme difficulty to forecast its performance by analytical methods.

Experimental work to determine fuel injection characteristics has followed two main avenues. In the first procedure, that followed by Juhasz[6], Schweitzer[7], Davies and Giffen[4] and other workers, the oil discharge rate was accurately measured throughout the injection by the now well established slotted disc method.

The disc works on the stroboscopic principle. A slot of definite width in a disc driven at pump shaft speed cuts the path of the oil jet as it leaves the nozzle and the fuel ejected while the slot passes the nozzle is collected and measured. The phase angle at which fuel is collected may be varied over the whole of the period of injection so enabling a curve of rate of injection per degree of pump angle to be plotted against pump-angle. When converted to a curve of nozzle velocity against time, it becomes possible, knowing the characteristics of the nozzle, to determine the pressure at the nozzle.

Investigations on the lines set out above especially if supplemented by a study of the form and distribution of the spray as carried out for instance by Juhaz (8) provide information which is invaluable in the analysis of subsequent engine performance.

The second method of attack which should logically form the complement to the foregoing employs a method of pressure measurement at two points in the fuel line. The first pressure measurement is taken as near to the pump outlet as is practicable and the second close to the nozzle. By this means, it becomes possible to correlate pump action and nozzle performance, to determine the hydraulic conditions obtaining in the system and to interpret engine performance in so far as it may be influenced by such conditions.

It was by this latter method of attack that the author decided to make the investigations. At the outset, it was realised that a radical departure from the recognised methods of pressure measurement would be necessary to register accurately the large and rapid pressure fluctuations which occur in the fuel line, and in the process of such measurement to make negligible the variation in the normal volume of the system. On the score of inertia and volume displacement, even the lightest piston and spring indicator was evidently unsuitable, thus restricting the possible use of existing instruments to the "Farnboro"[9] or "Hopkinson"[10] types or to modifications of these types.

Certain work on the lines of that proposed to be undertaken had been carried out by several investigators using the above instruments suitably modified. Farmer and Alock[5] in a paper read before "The Institution of Mechanical Engineers" describe briefly the use of the "Farnboro" Indicator for fuel line measurements. In both the paper and subsequent discussion, it is noted that this type of indicator is liable to give distorted results when subjected to violent pressure surges and that the pulsations of the indicator valve, causing it to act in the capacity of a hydraulic accumulator, upset the normal working of the injection system.

Le-Mesurier and Stansfield[11] who also made use of this instrument for fuel line measurements note that the recorded pressure is influenced by the friction of the moving parts and that this friction varies with the viscosity of the fuel used. They also note that there is an appreciable time lag due to inertia. An optical type of indicator has been used by Berg and Rode[12] in an extensive series of tests carried out in the experimental department of the Deutche Werke Kiel A.G. In these experiments, a stiff diaphragm exposed to the oil, communicates the pressure variation through small movements to a mirror which reflects a fine spot of light on to a rotating photographic film. While the system adopted, judging from the published results, has considerable merit, it is evident that the necessary rigidity of the indicator limits accurate registration to the higher pressure variations.

From the experiments of the above investigators, it would appear that the movement necessary to indicate pressure change must be exceedingly small, and the inertia of the intermediate linkage negligible. The only form of coupling which will meet the exacting condition is electrical in nature, and such a coupling is found ideally in the Cathode Ray Tube.

In this tube, an electron jet is produced from an oxide coated cathode and after passing between two pairs of mutually perpendicular deflecting plates, the focussed beam strikes a fluorescent screen giving a luminous spot. The beam may be deflected by suitable voltages applied to the deflecting plates. The problem of indicating the oil line therefore reduces to the conversion of pressure change and pump motion in-to two alternating potentials and applying these in their correct phase to the Cathode Ray Tube.

At the commencement of the investigation, little published work on methods for effecting such conversion was available, and the final decision was largely based on theoretical considerations of established principles.

A broad basis of differentiation of pressure sensitive devices suitable for oil-line pressure indicating may at once be made in accordance with the nature of the separator isolating them from the working medium. A diaphragm is normally employed for this purpose, and pressure conversion devices fall into two classes depending on whether a light or rigid diaphragm is employed. A light diaphragm only fulfils the function of a separator and the pressure sensitive element must withstand the loads imposed. On the other hand, a rigid diaphragm resists the pressures imposed, the resulting distortion being utilised to indicate the pressure change.

A brief account of the principles considered is given under (I) Light Diaphragm and (II) Rigid Diaphragm.

(I) LIGHT DIAPHRAGM. Piezo Electricity.

In 1880, F. and P. Curie[13] discovered that certain crystals which were known to generate electric charges when heated or cooled also developed similar polarisation when subjected to stress, and that the charge induced was directly proportional to the pressure. Little use was made of the phenomena until this century when Professor Cady[14] and others introduced piezo electric crystals for the control of high frequency oscillators. Many substances are known to exhibit the piezo electric effect, the best known being Rochelle Salt, Quartz, and Tourmaline.

In 1919, Sir J.J. Thompson[15] suggested that as the quantity of electricity generated is directly proportional to the pressure, the pressure of explosions might be gauged by direct measurement of the quantity of electricity produced by crystals subjected to the pressure. He proposed the use of the Cathode Ray Tube in conjunction with crystals of tourmaline and this method was later employed by Keys[16].

The factors which seemed to make such a method particularly applicable to the proposed work were:

(a) The employment of a light diaphragm might be expected to give a more sensitive indication of small pressure variations than would a rigid diaphragm.

(b) Suitable proportioning of crystals and diaphragm should give sufficient charge to operate the Cathode Ray Tube with simple or no amplification.

(c) Since output is directly proportional to the pressure, no electrical integration is necessary.

(d) The deformation of quartz or tourmaline crystals even under high pressures would be so small as to cause negligible increase in the normal volume of the injection system.

On the other hand, the input impedance of the system is high, a disadvantage which is however fairly easily overcome in a laboratory instrument. Again, the extent to which the output of the crystals might be affected by temperature variation, atmospheric conditions and pollution was problematical.

(2) RIGID DIAPHRAGM.

(a) Resistance Pile. When compressed, a pile of resistance discs of carbon or similar material suffers a change in resistance and is thus able toproduce a current change which in turn will produce a voltage swing across a series resistor. The change of resistance under compression is largely a surface effect, and it was considered that the possible effects of atmospheric change on the operation of the piezo electric system could only be magnified in the case of the resistance pile. The change of resistance is only proportional to change of pressure over a very small range, and the system is liable to reach a saturated state where increase of pressure produces no further variation in resistance. Using carbon or similar material with low elasticity, the system is also subject to hysteresis effects.

(b) Capacity. A rigid diaphragm forming one of the plates of a condenser may, through its motion, be made to vary the capacity of an electrical system and thus create a voltage swing across a series resistor. The E.M.F. is proportional to the acceleration of the moving diaphragm and thus requires electrical integration to give a pressure-time diagram.

Owing to the necessity of keeping the volume change of the fuel line as small as possible, the condenser would necessarily be of small dimensions and thus the voltage swing would be small, requiring considerable amplification. Calibration of the system would also be a matter of some difficulty. As an alternative circuit, the insulated plate of the condenser might be coupled to the grid of an electrometer triode followed by suitable D.C. amplification, but if the instrument were to be used as a universal indicator, it would probably be necessary to provide for a variable condenser gap as, with the fine clearances necessary, a gap suitable for high pressures would be unsuitable for lower pressures.

This alternative circuit has recently been adopted in the Metrovick-Dodds indicator[17] for measurement of spray valve lift and it has been suggested as a suitable method for the measurement of cyclic vibrations in engines.

(c) Magnetic. If the poles of a permanent magnet are separated from a rigid diaphragm by a few thousandths of an inch, the diaphragm vibrating in conformity with pressure changes in the system will vary the gap between it and the magnet and give rise to voltage variations in coils wound on the magnet poles. Since these voltages are determined by the velocity of the diaphragm, they are proportional to the rate of change of pressure in the system indicated and therefore require electrical integration to give a true pressure-time diagram.

A very satisfactory aspect of this system in its low input impedance which is in the order of 1000 ohms, and therefore no special precautions as to insulation or capacity of the leads from the magnetic pick-up need be taken.

(d) Photo-electric. If a rigid diaphragm be highly polished and light from a point source directed thereon the intensity of the reflected light as received by a photo-electric cell will be proportional to the curvature of the diaphragm, and the current in the cell will vary accordingly. It thus becomes possible with simple D.C. amplification to obtain a direct pressure-time relation.

(e) Conductivity. If a fixed insulated electrode immersed in a fluid of low conductivity be set in close proximity to a rigid diaphragm separating it from the oil in the fuel line and acting as a moveable electrode, the movements of the diaphragm conforming to variations of pressure will vary the length of conducting path, and hence the resistance to an electric current passing between the electrodes. A fixed resistor placed in series with this variable resistance will serve to produce a voltage change which may be impressed on the grid of a suitable amplifier, and the amplified voltage change led directly or through a suitable condenser to the plates of the Cathode Ray Tube.

Possible contamination of the electrode surfaces and variations in the properties of the conducting fluid are questions which would require careful consideration in the adoption of such a system for pressure measurement, but due to the facility with which these factors could be guarded against the principle has been applied in the present research to the measurement of the spray valve lift.

After full consideration of the principles outlined above, the author came to the conclusion that the advantages inherent in the use of a light disc in conjunction with piezo-electric crystals, including ability to transmit the finer pressure variations, and linearity of response justified the adoption of this system. Several indicators notably the Cossor[18], Metrovick-Dodds[17] and Standard-Sunbury[19], based on principles other than that adopted have since been evolved, and their operating characteristics, together with those of a piezo-electric system used by Rowe in the University of London, have been described in an informal discussion held by the Diesel Engine Users' Association.[20].

In the comparison of the various types the piezo-electric system apart from its admittedly desirable features of non-saturation and linear response appeared to suffer on the score of fragility and liability to error through electrical leakage.

On the question of fragility, the author can state that the original crystals in the present research have been used continuously with only occasional removal for inspection over a period of three years, and have withstood hundreds of hours of work often under severe conditions of speed and loading. The electrical leakage from the crystals and their associated amplifiers is however, a more disturbing factor. The meticulous care and cleanliness necessary to maintain a time-constant in the order of 30 minutes as required by Rowe[21] for his work on the effects of pressure release on injection almost precludes the use of the piezo-electric system for an extended series of observations.

In the present series of experiments, the extreme period of the observed phenomena is about one fortieth of a second and the time constant, observed as $^1/_5$ second, is considered adequate. To confirm this point, the delivery valve from the pump was removed to ensure that atmospheric pressure would obtain in the pipe on the cessation of delivery and photographic observations of the pressure at the pump and nozzle were made at low pump speed and large charge setting. Electrical leakage would be indicated by a drop in pressure

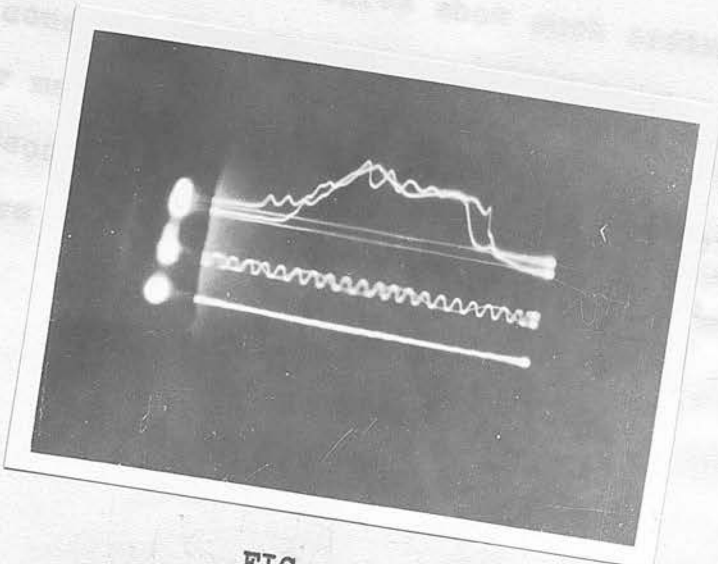


FIG. 3.

below atmospheric and as may be noted from the records shown in Fig. 3 is of very small magnitude. To obtain an atmospheric pressure-base, it was assumed that the pressure at the measuring point near the pump would momentarily be reduced to approximately atmospheric when the pump plunger registered with the suction part. Careful measurement of the ordinates between this assumed atmospheric base and the lift point of the needle valve show such ordinates to be sensibly constant for similar speed and load conditions, and, after making allowance for the inertia of the needle, and its associated equipment, to be equal, according to the pressure scale, to the particular injection setting of the needle.

# PART III.

## DESCRIPTION of APPARATUS
## and
## PHOTOGRAPHIC RECORDING.

A. Mechanical Details.

B. Selection, Preparation and Mounting of Crystals.

C. Crystal Output.

D. The Cathode Ray Oscillograph.

E. Associated Equipment.

PHOTOGRAPHIC RECORDING.

## PART III. DESCRIPTION OF APPARATUS.

### SECTION A : Mechanical Details:

Pump (Fig. I). The pump used is a standard four-cylinder Bosch No. P.E. 4075/201 fitted with a 7 mm. plunger and standard Bosch type delivery valve. As only one cylinder was utilised, the delivery valves of the remaining three were removed, and their deliveries by-passed to the pump suction through a breeches-piece connecting their discharge ports. The pump plunger barrel and delivery valve were taken new from stock at the commencement of the experimental work.

Speed Control. Originally, the pump was direct coupled to a 1/2 H.P. 1650 r.p.m. 220 Volt. D.C. shunt wound motor with constant field excitation and a regulating resistance in series with the armature. An induction motor-generator set of 1.5 kW output at 220 volts was used to supply power to the fuel-pump motor. The method used of controlling the speed of this motor was inconvenient, inasmuch as the inherent series characteristic imparted by the armature resistance made the speed dependent almost entirely on the load (or throttle opening with the result that adjustments had to be made to the rheostat for every alteration of throttle position. In addition, several other disturbing influences such as the gradual heating up of the regulating resistance and of the motor itself made the speed gradually creep from the initial set value.

To overcome these drawbacks and at the same time to utilise existing equipment, it was decided to use the Ward-Leonard system of speed control with separately excited fields for both generator and motor, the armatures of the two machines being permanently connected together. The necessary excitation power was obtained from a 350 Watt 230 Volt Compound Wound Generator mounted adjacent to the motor-generator set and driven off the motor spindle by means of V belt and pulleys.

The shunt field of the main generator was excited from this auxiliary machine through a set of five resistances, consisting of 15, 15, 20 and 25 Watt 250 volt metal filament lamps each controlled by a separate switch and all wired in parallel. By manipulating these switches in different combinations it was possible to obtain eleven definite excitations, and owing to the extremely small thermal capacity of the lamps, steady excitation conditions for each setting were reached almost immediately. Since the field circuit of the fuel pump motor was supplied at a constant 230 volts, it was possible to obtain eleven different speeds proportional to the e.m.f. of the main generator induced by the various excitations. These available speeds ranged from approximately 200 to 1200 r.p.m. but in the course of the investigations only five convenient speeds ranging between approximately 200 and 950 r.p.m. were used. To improve the flatness of the load speed characteristic of the fuel pump motor over the working range, advantage was taken of the series field winding on the main generator.

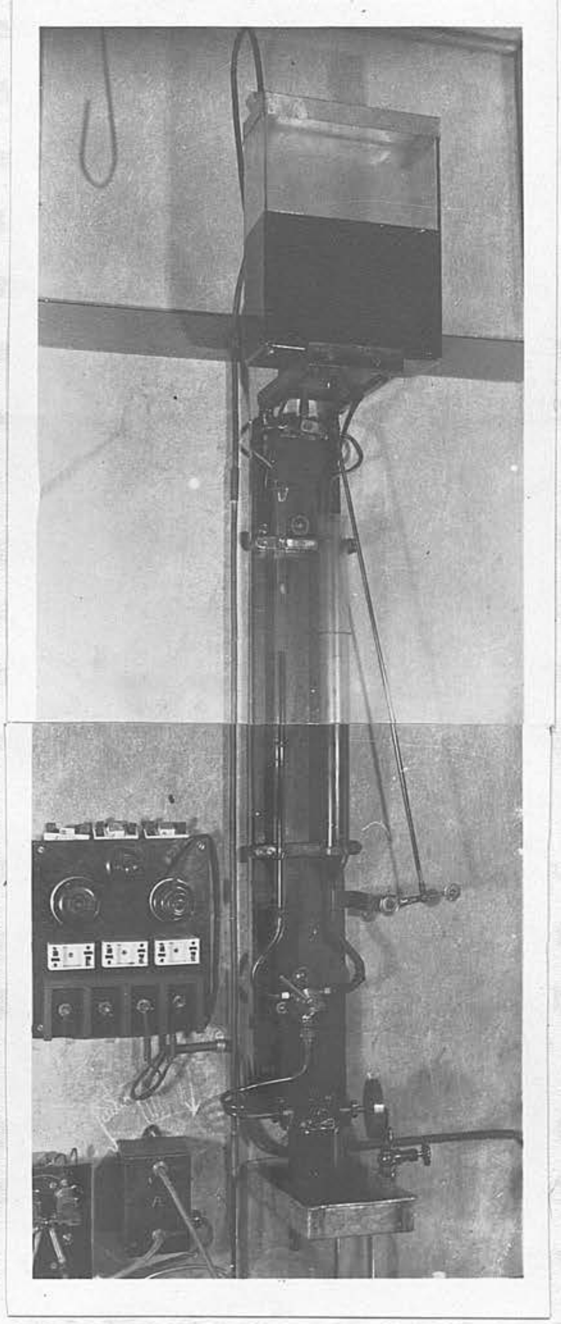


FIG. 4.

FUEL SUPPLY.

By permanently connecting this with the armature, it was possible to obtain an increasing generator e.m.f. with increasing load, and this increased e.m.f. tending to compensate for the resistance drop in the motor armature maintained a more constant speed on varying loads.

The drop in pump speed between 1/4 and 3/4 throttle opening was only about 10 r.p.m. This gave satisfactory and consistent results and the whole of the investigations were carried out under these conditions.

Oil Supply. (Fig. 4). Oil is supplied from a glass reservoir fixed some six feet above the pump suction. Through a two-way cock, the oil is fed at will to a graduated burette or to a large glass cylinder, these terminating in the common outlet of another two way cock, from which the oil is led through a Bosch Filter to the pump suction. Such an arrangement allows of instant change-over being made from the large glass working cylinder to the small graduated burette for the purpose of making discharge measurements.

Charge Control. (Fig. 5). Charge Control is by a sliding rack gearing with pinions clamped to slotted sleeves engaging with the pump plungers. The full travel of the rack is 24 millimetres and for the experimental work, this range was divided into four equal divisions designated $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$ and full charge respectively. The charge settings are obtained

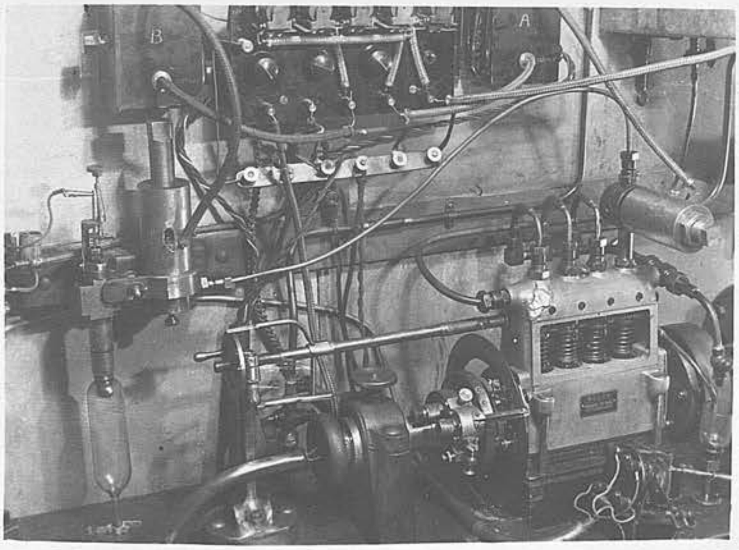



FIG. 5.

CHARGE CONTROL.

by a micrometer wheel screwed 24 threads per inch operating a spindle similarly screwed and connected by fork and fitted bolt to the control rack. The edge of the micrometer wheel is divided into five equal parts, these in turn being sub-divided into tenths, and works against a scale fixed to the screwed spindle. A spring mounted in the pump housing maintains pressure between pinion and rack and serves to take up backlash between these two members.

Choice of Crystals. Crystals of tourmaline and quartz possessing relatively high piezo-electric constants appeared to be the most suitable and the former mineral was chosen in preference to the latter for the following reasons:-

(a) Black Tourmaline has a fairly wide distribution in Western Australia and particularly good specimens are to be found near Greenbushes in the south-west of the State associated with the tin-bearing lodes occurring in that district. Although the black variety has a lower piezo-electric constant, than the clear gem varieties found in Ceylon and Brazil, it was found on test to be adequate.

(b) Quartz crystals are subject to twinning, a defect which nullifies the piezo-electric effect. Tourmaline, on the other hand, is remarkably free from this defect.

(c) The electrical axis in Tourmaline coincides with the optical or Z axis whereas in Quartz, the electrical axis is perpendicular to the optical axis and is not an axis of symmetry. This feature of Tourmaline makes for ease in preparation and often makes possible the cutting of a satisfactory piezo-element from a somewhat imperfect specimen.

SECTION B. CUTTING and TESTING.

Crystal Selection, Preparation and Mounting.

From selected black Tourmaline Crystals, slabs about ¾ inches thick were cut at right angles to the optical axis by means of a copper rock saw fed with No. 80 carborundum powder. The slabs were then placed inside a steel ring mounted on a flat steel plate, and secured by bitumen poured hot into the spaces between crystal and ring. Cores 5/8 inches diameter were cut by means of a copper tube, 11/16 inches inside diameter rotated by a high speed drive and their ends ground parallel to within 1/1000 inch on a zinc rock-sectioning table.

For making rough tests of the piezo-electric effect of the cores, a small handpress was made from a 3 inch valve cover and spindle. In the gland recess, a block of marble was fitted and the end of the valve spindle recessed to take a half ball. The crystal was compressed between the faced half ball and a thin steel plate resting on the marble block, and the electrical charge developed on release of pressure measured by a valve voltmeter, consisting of a Cosser P.I. Triode with a series filament resistance, 12 volts in the Anode and 0.15 milliammeter in the plate circuit.

Sudden release of pressure will either increase the plate current or completely suppress it depending on which face of the crystal is connected to the grid. On first using this simple testing device, widely varying results were obtained even from cores cut from the same crystal slab. These were found to be due in large measure to variations in resistance of the cores caused possibly by inclusions of mineral oxides and deposits of cutting and polishing compounds. After boiling for 15 minutes in strong hydrochloric acid, followed by repeated boiling in water and alcohol and careful drying, resistances rose to infinity in all cases.

Grinding and Polishing. To prevent fracture under load and to ensure uniform distribution of pressure, it is essential that the crystal faces be made flat. Prolonged hand grinding with fine powder on surface plates tends to produce a domed surface due to the breaking up of the cutting material at the edges and resultant reduced abrasive effect at the centre of the specimen. To overcome this effect, the crystal faces were first made slightly concave by a copper lap and then ground until the concavity had been removed.

Following the rough grinding, the faces were hand-ground and polished on glass, using the finest emeries, and the flatness gauged by using the principle of light interference. When the air film between the crystal face and optically flat glass is viewed in sodium light, the alternate interference and reinforcement of the monochromatic sodium rays reflected from the upper and lower boundaries of the film show as a series of alternating light and dark bands. If the boundaries of the air film are flat, but in the form of a wedge, the bands will appear as par llel lines and within the range of an adjacent light and dark band, the thickness of the wedge has altered by an amount equal to the wave length of the light employed. On the other hand, if one of the boundaries is plane and the other curved, a pattern of concentric rings is formed which may be looked upon as a contour of the curved surface, the contour height being the wave length of the light used. On first lightly placing the glass on the crystal face, a series of curved, roughly parallel bands are seen when viewed under oblique sodium light (Fig. 6), but with the application of pressure between glass and crystal, the air is slowly squeezed out from between the surfaces and the interference pattern takes the form of a series of rings forming a contour of the curved crystal face (Fig. 7). It thus becomes possible to rapidly and accurately determine the progress of polishing the crystal face. Records of progress at various stages are shown in Figs. 7 - 9, and using the technique mentioned previously of first making the face concave and removing the concavity by grinding, little difficulty was experienced in obtaining optically flat crystal faces. It was found advisable however, in practical application to leave the crystals slightly domed in order to relieve the stress at the outer edges.

After grinding and polishing, the crystals were tested between flat plates on the Brinell Hardness Machine

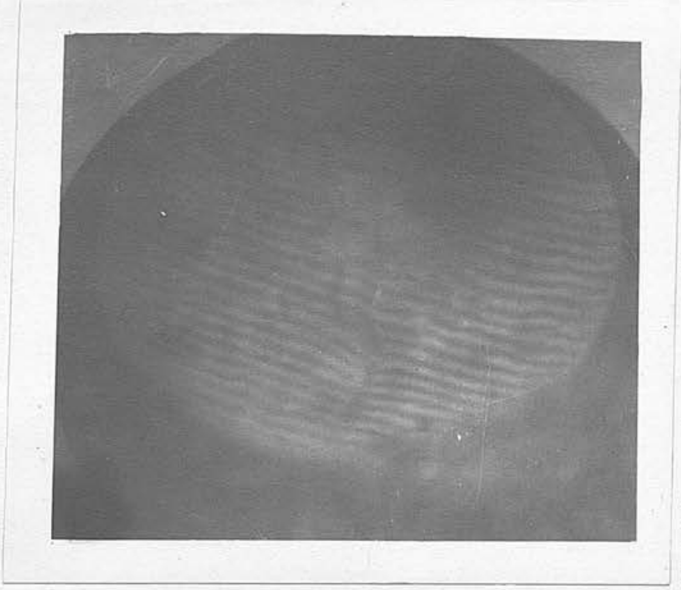


FIG. 6.

INTERFERENCE PATTERN.

due to

AIR WEDGE.

FIG. 7.

FIRST STAGE of GRINDING.

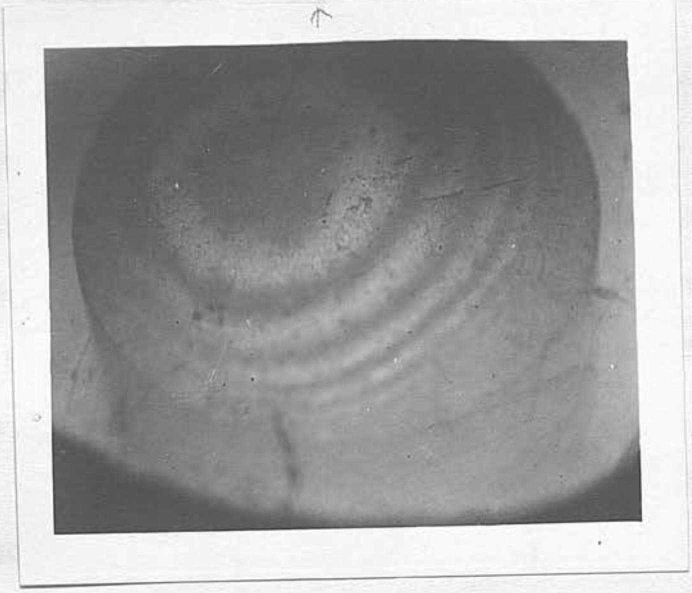


FIG. 8.

SECOND STAGE of GRINDING.

FIG. 9.

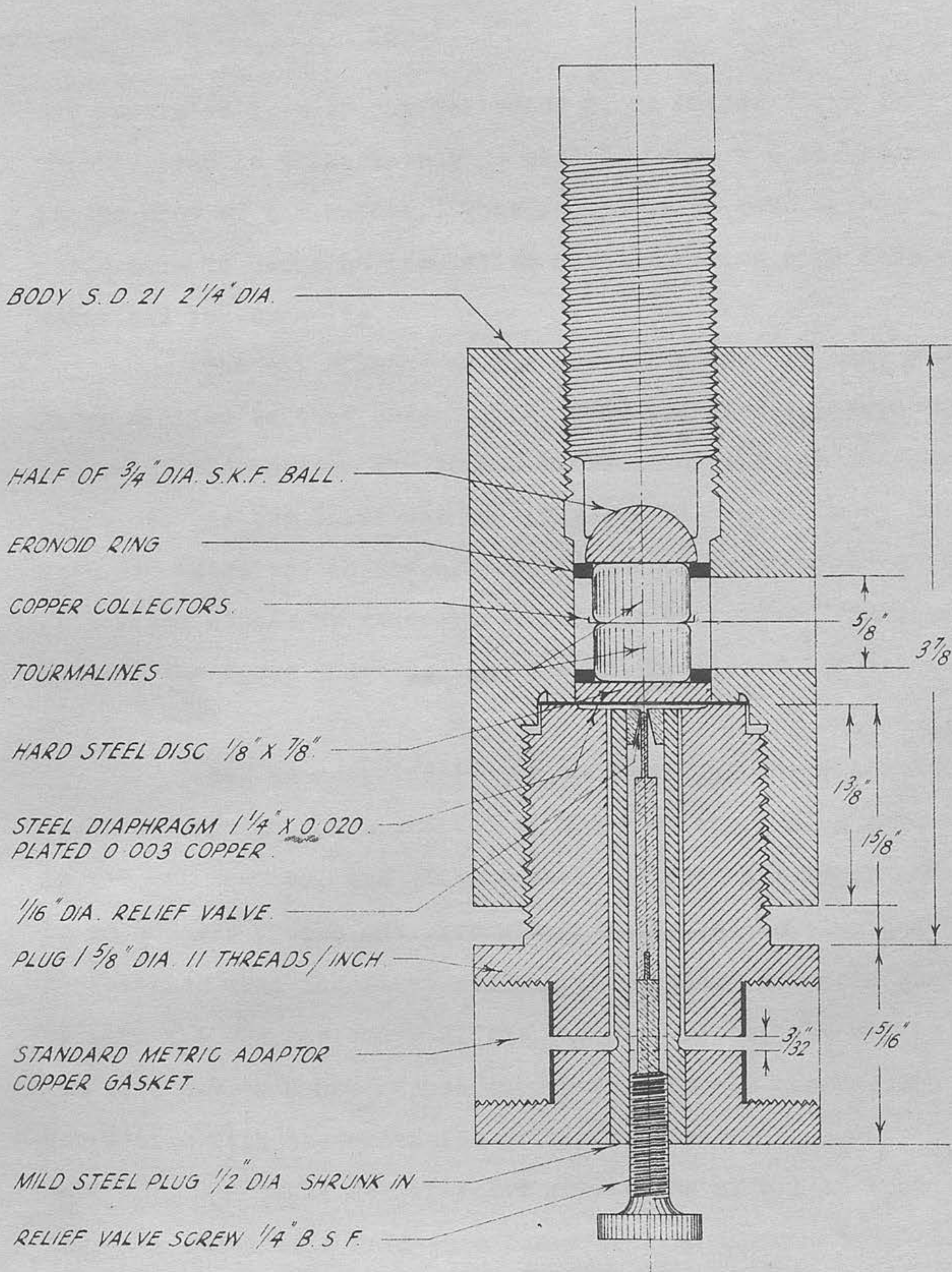
FINAL STAGE of GRINDING.

and withstood the maximum load of 5000 kilograms without the least sign of fracture. Under the practical working conditions of the investigations, the load on the crystals with a maximum pressure of 5000 lbs. per square inch in the fuel lines is only some 3000 pounds. The safety factor is therefore of a reasonably high order.

<u>Crystal Holder</u>. To minimise deformation, the crystal holder was made exceptionally rugged and the distortion and consequent change in capacity of the system when under load is negligible. The holder is machined from a 65 ton nickel steel to dimensions and form shown in Figs. 10 and 11.

Referring to Fig. 10, the crystals are located in the body by Eronoid rings fixed near the earthed faces, thus affording a certain flexibility of crystal movement, at the same time maintaining an insulating air space between the insulated crystal faces and the steel body. The lower crystal face bears on a $\frac{1}{8}$ inch disc of hardened ground and polished steel resting on the diaphragm and fitting easily in the body of the holder, while the upper crystal face contacts with a faced steel ball located in the hardened recess of a screwed steel plug.

With this system of location, even contact between all the bearing surfaces of the crystals is assured and irregularities in loading accommodated. In operation, the positive crystal faces are earthed and the negative faces




CRYSTAL HOLDER
SCALE: FULL SIZE

FIG.10

are separated by a copper collector 0.004 inches thick from which a lead is taken through a slot $\frac{3}{8}$ inches x $\frac{3}{8}$ inches cut in the body of the holder. This procedure is considered perferable to using an insulating bush as giving high insulation and low capacity.

The oil chamber of the crystal holder is made from steel similar to that used for the body and the diaphragm is securely held between the lapped surfaces of the oil chamber and body. As the joint has to withstand high pressures, great care was exercised in its preparation, and its accuracy checked at various stages by squeezing lead wire of uniform diameter between the faces and measuring the uniformity of its deformation.

Oil is admitted to the underside of the diaphragm through longitudinal slots cut in a $\frac{1}{2}$ inch steel plug shrunk in the oil chamber, the slots connecting at their lower end in an annular groove cut circumferentially in the plug and registering with the inlet and outlet standard adaptors screwed into the chamber body. From the plug slots, the oil is led to a cavity 0.002 inches deep formed in the chamber and connecting with the underside of the diaphragm.

A small relief valve consisting of a 1/16 inch diameter steel ball held in a brass seating screwed into the plug, serves to bleed the air from the system.

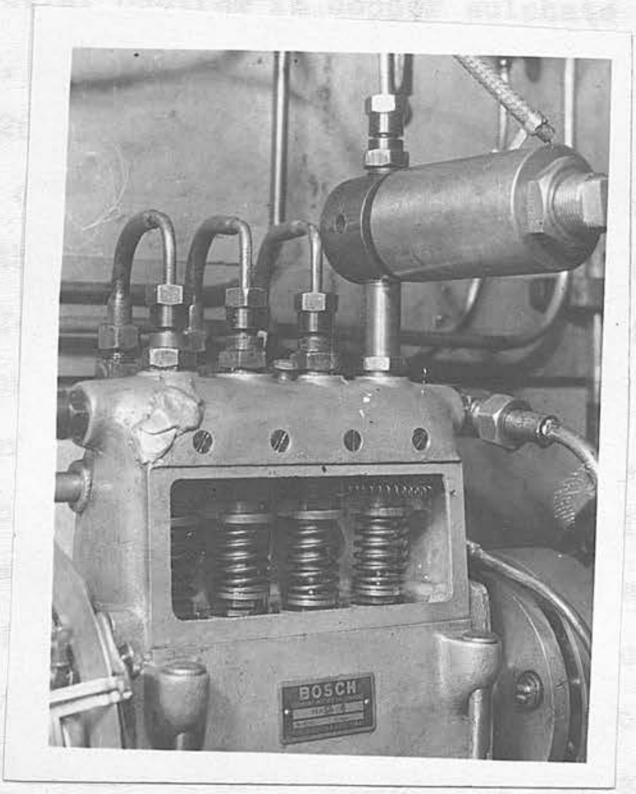



**FIG. 11**

**CRYSTAL HOLDER.**

**(Pump End).**

<u>Diaphragms</u>. The diaphragms of tempered steel were accurately ground on glass to an even thickness of 0.020 ins.

In order to ensure a perfect oil seal, the diaphragms were plated with copper. A thin coating in a cyanide bath was followed by a heavier coating in copper sulphate solution and after regrinding, the finished size gave a total diaphragm thickness of 0.026 inches made up of 0.020 inches steel with 0.003 inches of copper on each face. The added capacity to the system with the introduction of the holders is, as will be observed from Fig.37 of a small order and the variation of capacity during pressure measurement negligible.

The crystal holder near the pump (Fig. 11) is connected to the pump discharge through a short distance-piece conforming in dimensions to the standard fitting at the pump connection and drilled 3/64 inches thereafter to connect pump discharge to crystal holder. At the nozzle end of the pipe, the second crystal holder (Fig. 12) in every way similar to the holder described above, is connected directly to the nozzle holder by a standard adaptor. A general view of the unassembled components of the crystal holder is shown in Fig. 13.

## <u>SECTION C</u>. <u>CRYSTAL OUTPUT and its MEASUREMENT</u>.

While the impedance of the Cathode Ray Tube is extremely high, it does nevertheless seriously affect the output from a source such as the crystals used in the present

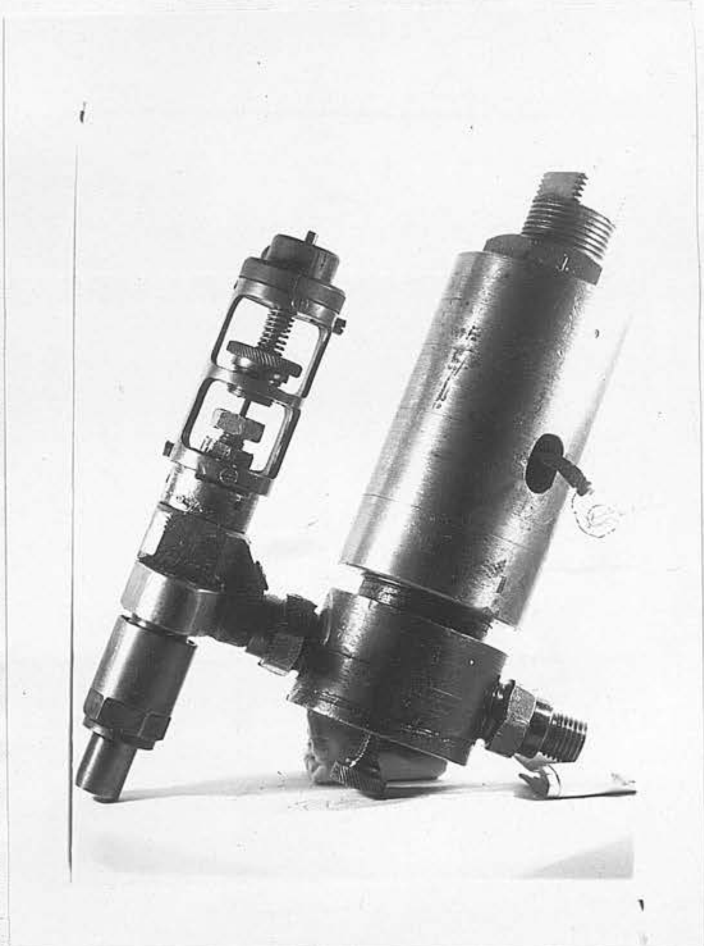


FIG. 12

CRYSTAL HOLDER

(Nozzle End)

FIG. 13.

COMPONENTS of PIEZO PILE.

investigations. With the crystals directly connected to the Oscillograph, the rate of leakage is slightly less than the rate of generation of charge, and capacity remaining constant, the voltage will be proportional to the rate of generation of the charge. The resulting record will therefore show the rate of change of pressure or in other words a "velocity" diagram.

An enlargement of such a diagram obtained with direct connection of the crystals to the tube is shown in the lower diagram of Fig. 14. The upper diagram in the same Figure is a Pressure-Time record taken under exactly similar conditions, and, when graphically differentiated with scale suitably altered, the resulting curve fits the "velocity" diagram with remarkable accuracy. The usefulness of a velocity diagram in exaggerating small pressure changes is well brought out in a comparison of the two records.

The problem of recording the output from the crystals is similar to that of the delineation of the wave form of atmospherics investigated by the Radio Research Board[22]. Briefly, the solution consists in absorbing the charge produced on a condenser thus limiting the voltage swing to an order suitable for application to the first grid of a D.C. amplifier. The valve employed in the first stage of the amplifier is an Electrometer Triode having exceptionally high input impedance followed by one or more stages of amplification. The Electro-

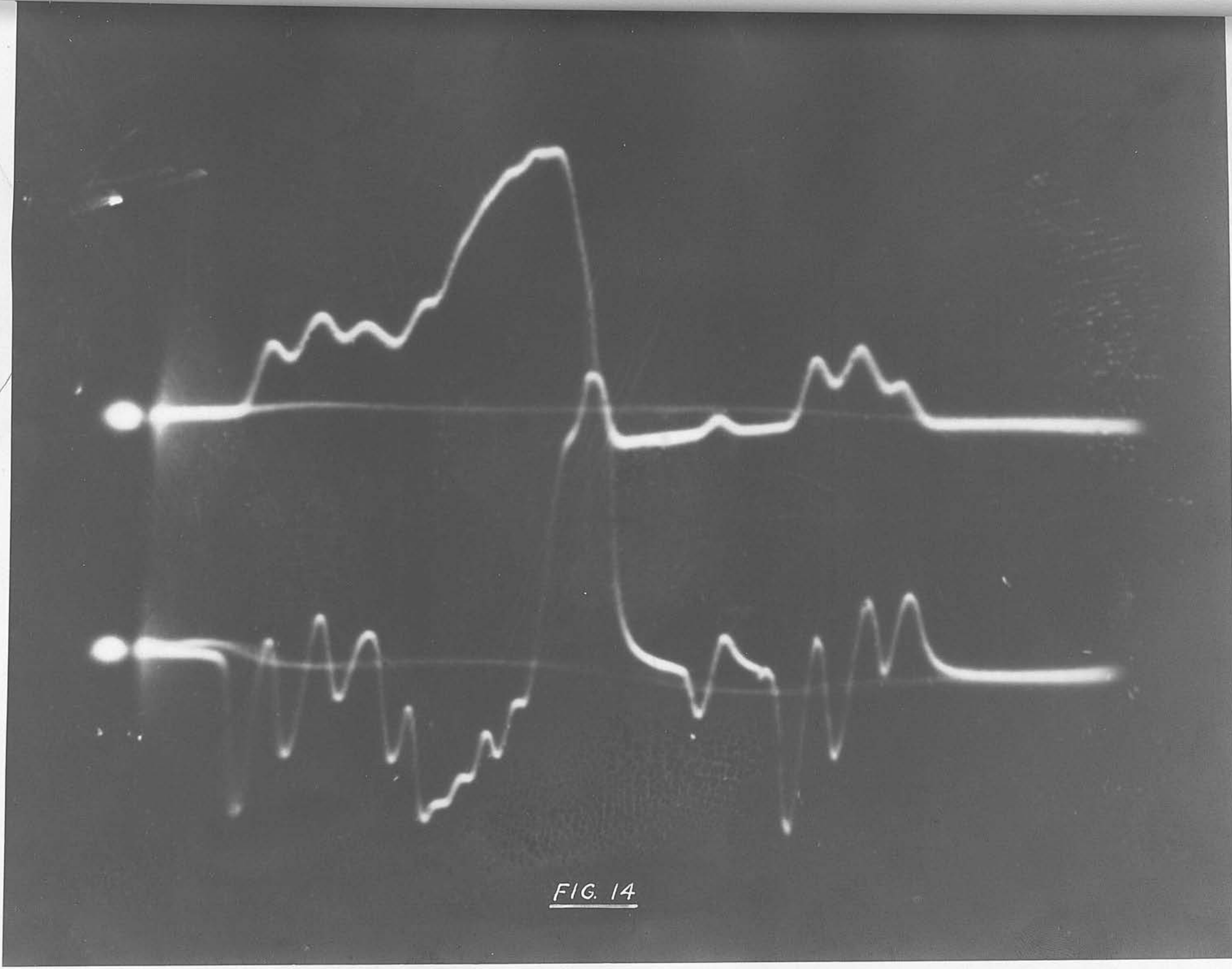


FIG. 14

meter valve is however prone to instability except under ideal working conditions and an alternative means of recording the crystal output was sought.

Following the work of Johnson and Nezert[23] on the measurement of small alternating voltages, an indirectly heated power pentode (R.C.A. 38) was found to give satisfactory results. Using 4 volts on the heater, 12 volts on the anode, and 6 volts on the screen grid, the valve functions as an electrometer with a grid resistance calculated from exponential decay curves of over 500 megohms. Further experiments showed that by using higher anode voltages sufficient modulation could be obtained to give a satisfactory record on the Cathode Ray Tube without the necessity of further amplification and without reducing the output impedance to dangerous limits. As finally applied, the valve employs 4 volts on the heater, 90 volts on the anode, and 60 volts on the screen. The grid is biassed to the straight portion of the Grid-Volts-Anode-Current curve and the grid swing limited by a suitable condenser.

## SECTION D. THE CATHODE RAY OSCILLOGRAPH.

General Arrangement. The Oscillograph with its component apparatus is mounted as a self-contained mains-operated unit as shown in Figs. 15 and 16. The camera (Fig. 17) fitted with a Dallmeyer f1.9,3. lens and compur


FIG. 15.

FRONT VIEW.

CATHODE RAY OSCILLOGRAPH.

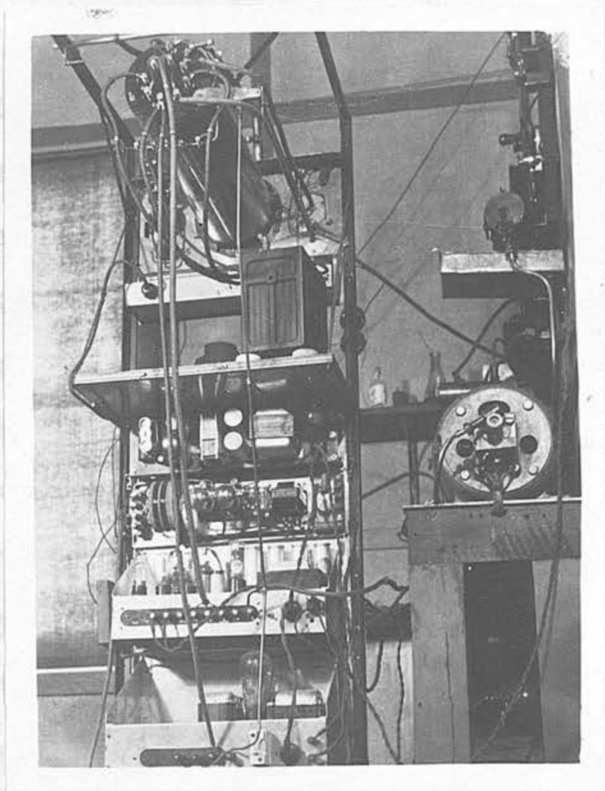


FIG. 16.

REAR VIEW

CATHODE RAY OSCILLOGRAPH.

shutter is mounted at the top of the rack in a manner permiting of universal focussing movement (Figs. 18 and 19). During photography, a felt bellows prevents entry of extraneous light.

Below the camera, the Cathode Ray Tube is mounted in a panel carrying a sheet-metal shielding cylinder, and connections from the tube are taken through a Cossor low capacity socket to the panels mounted below (Fig. 16). The gun voltage and current-meter is placed at the right of the Tube panel (Fig. 20) and two stand-off insulators connected by shielded wire of low capacity to the "work" (Py) and "sweep"(Px) terminals at the back of the tube are fixed to the left.

In descending order below the tube panel are fixed the filament control panel (Fig. 20) the amplifier power unit, the 1000 cycle oscillator, the sweep and shift control for controlling the diagram on the screen, and the high-tension power supply (Fig. 21). A short description of these components is given in the following section of the thesis.

## Description of Oscillograph Components.

Filament Control (Figs. 20 and 22): The filament controls were originally incorporated in the high tension panel but due to pick-up of small 40 c/s potentials due to leaking transformer flux, were later transferred to a separate panel. Incorporated in the same panel is the potentiometer, which controls the gun voltage by regulation of the

FIG. 17.

CAMERA.

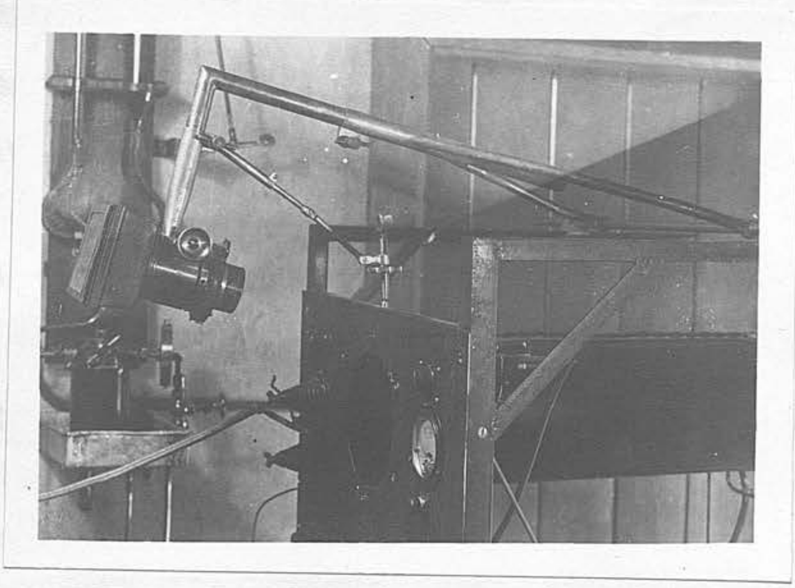


FIG. 18.

CAMERA RAISED.

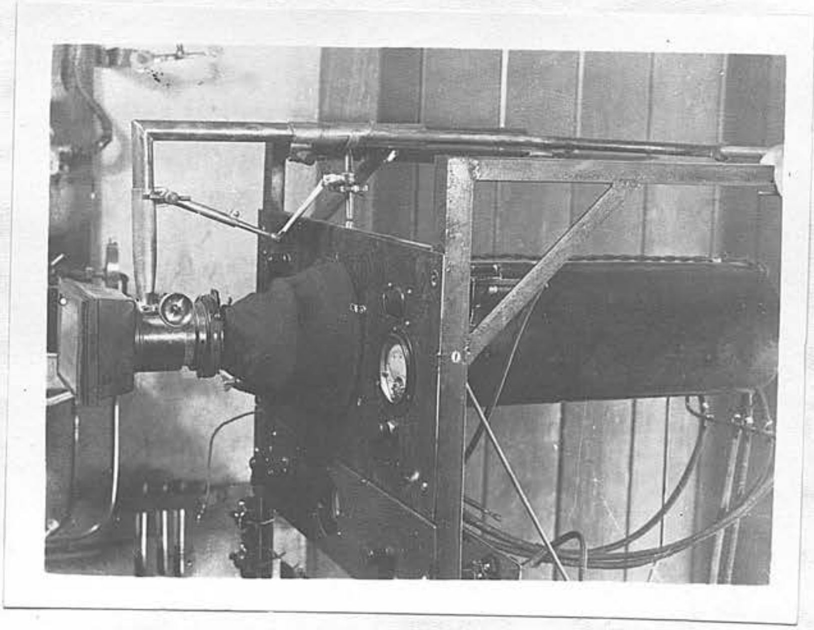


FIG. 19.

CAMERA LOWERED.

output from the mains transformer to the H.T. Transformer.

The shield focussing resistance gives a voltage drop in the positive lead of the gun circuit which is applied to a small shield surrounding the Cathode, and the electrostatic field produced serves to focus the electron jet before its acceleration from the gun.

Oscillograph Power Unit. The power unit employs a voltage doubling circuit as shown in Fig. 22. The primary of the mains transformer acts as an auto-transformer and in conjunction with a potentiometer gives two variable voltage ranges to the H.T. Transformer which supplies the rectifying valves of the voltage-doubler. A 0.25 megohm leak placed in the negative lead serves to throttle any large currents and improve regulation. The gun and shield of the tube are at earth potential thus serving to dispose of the stray space charges and electron jet current, and materially reducing the power required to drive the tube.

Sweep and Shift Current (Fig. 23). This circuit which was adapted from a circuit published in "The Wireless World"(24) employs hard valves in contradistinction to the more usual use of thyration tubes. The sweep provides a controllable rate of horizontal spot movement on the screen while the shift circuit determines the position of the spot.

A mains transformer supplies the voltages for the

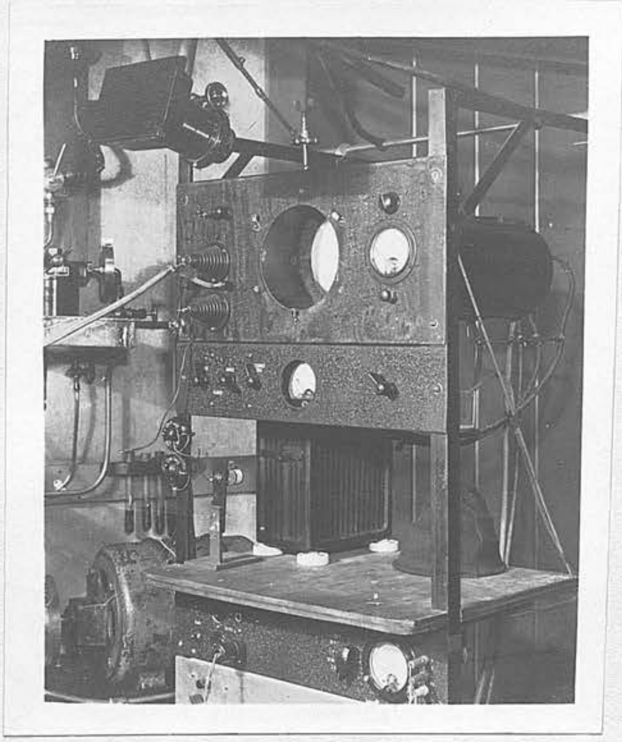


FIG. 20.

FILAMENT CONTROL PANEL.

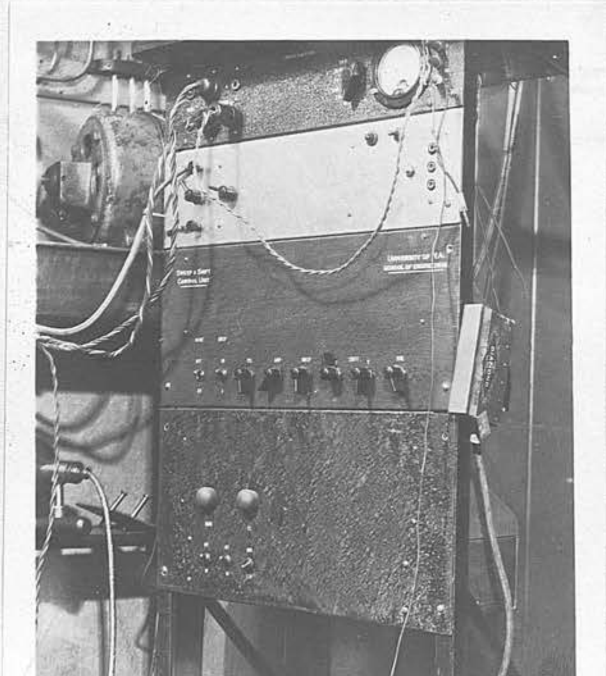
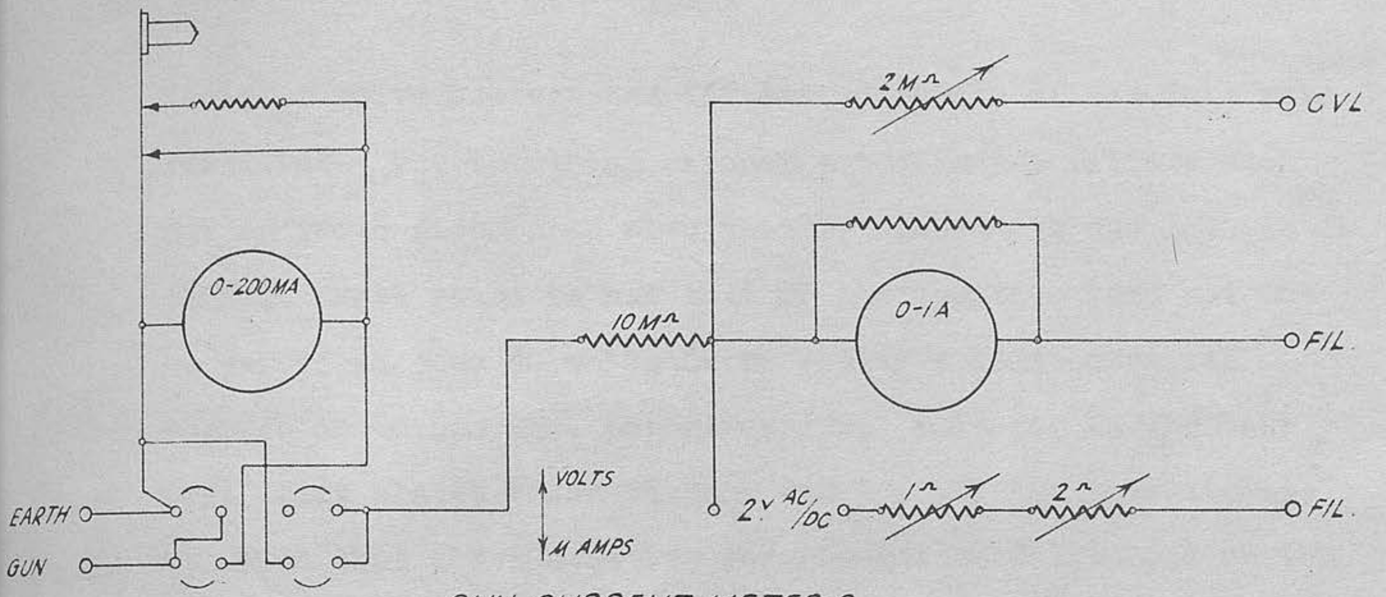
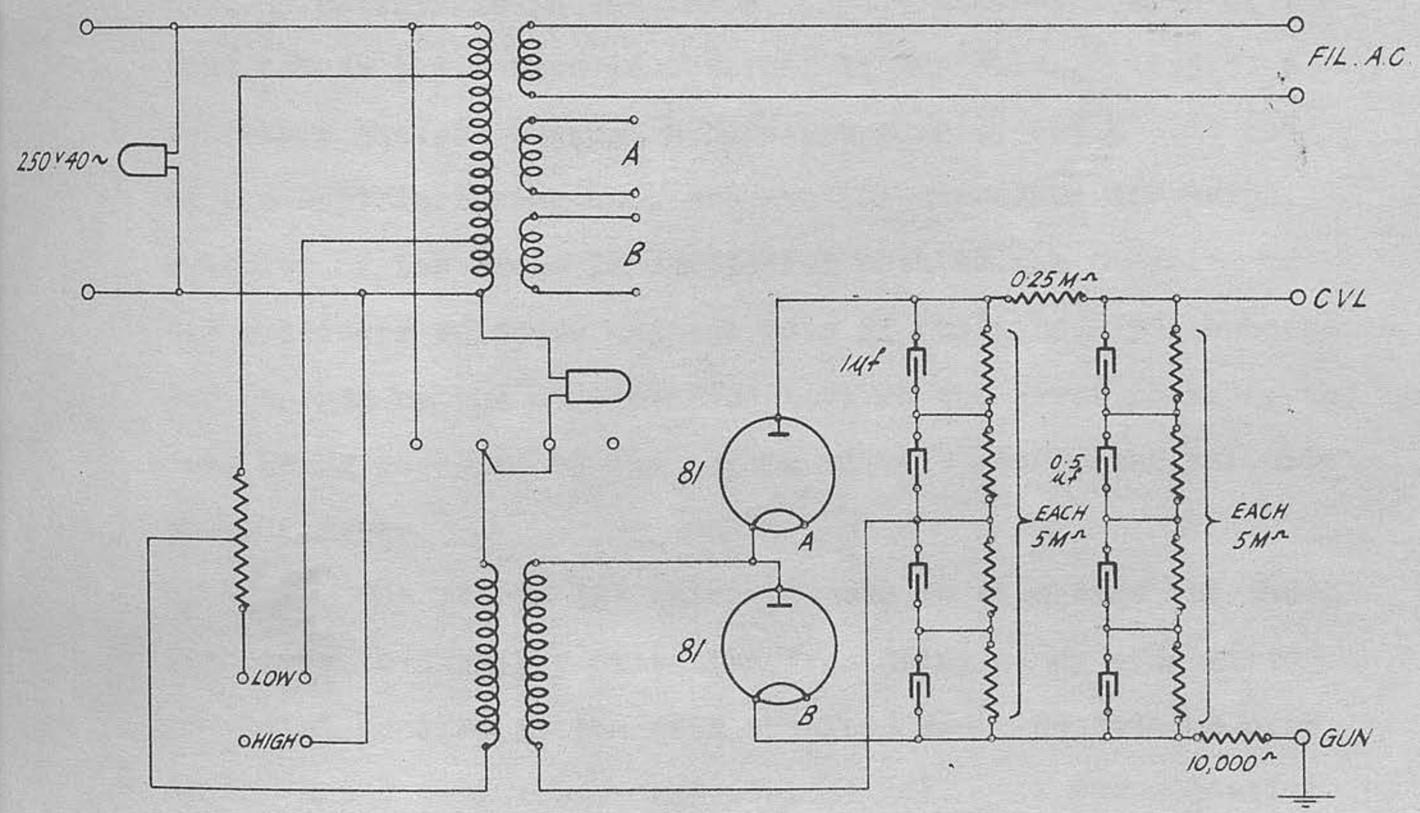


FIG. 21.




GUN CURRENT METER & FILAMENT CONTROL




OSCILLOGRAPH POWER PACK

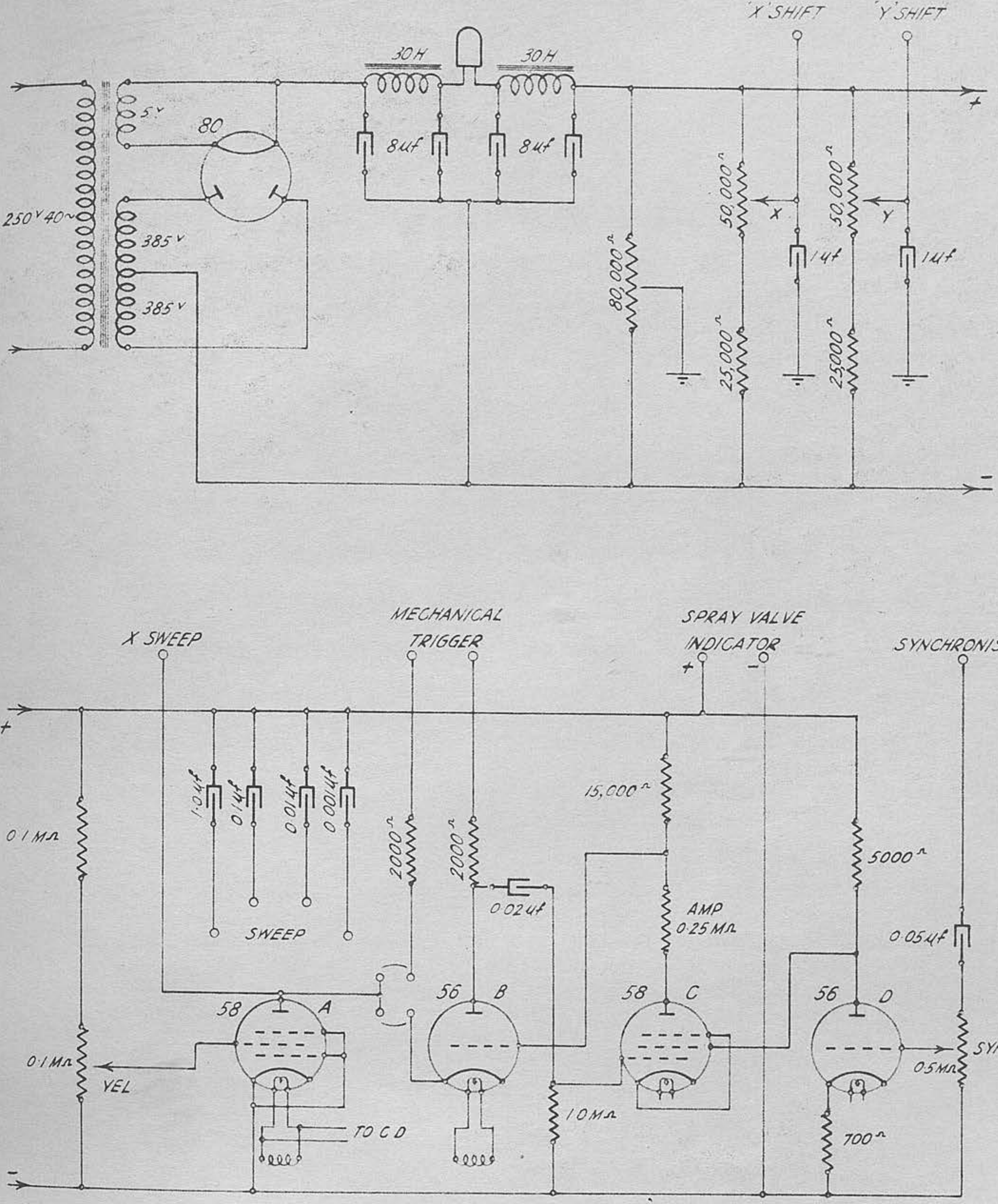
FIG. 22

indirect valve heaters and for the operation of the full wave rectifier. The smoothing circuit embodies two filters each having two 8 microfarad electrolytic condensers and a 30 Henry choke. Earth point is not that of the negative lead but may be varied to some 90 volts below positive lead potential through an 80,000 ohm. potentiometer. Voltages to the (px), (py) shift plates taken through two 50,000 ohm. potentiometers at 25000 ohms above negative potential enable the spot on the screen to be biassed to any desired position. The shifts are shunted to earth through a 1 microfarad condenser to eliminate pick-up potentials in the leads. Horizontal excursion of the spot across the screen is obtained by the voltage rise of a condenser charged through a constant current device, the return of the spot following upon exponential condenser discharge. Velocity of the sweep is controlled both by the capacity of the condenser selected and the rate of charging. The condensers are charged by the current flow through the first pentode, the rate being governed by the screen voltage set by the velocity potentiometer.

The Triode 'B' which is used to discharge the sweep condenser is normally prevented from doing so by a negative potential applied to the grid obtained from the voltage drop in the anode resistor of the pentode 'C'. If the potential across the condenser is sufficiently high however, current will flow through 'B' causing a potential drop across the anode resistor of valve 'B'. When applied to the control grid of the pentode 'C' current flow through the latter is prevented, thus removing the voltage drop across its anode resistor and thereby removing the negative bias on the grid of the triode 'B' with resulting condenser discharge.

The potentiometer controlling the amplitude is the anode resistor of valve 'C' and determines the point at which valve 'B' will allow discharge of the sweep condenser thus fixing the length of the sweep across the screen.

A triode 'D' is incorporated in the circuit to synchronise the sweep with the work voltage, a portion of the work voltage fed from the synchronising potentiometer to the grid of valve 'D' serving to lock a roughly adjusted sweep with the work. In the present investigations, an electromechanical trigger coupled to the pump shaft as described in the following section was used and synchronisation of sweep and work was therefore automatic. Through a switch incorporated in the lead from the anode of valve 'A' to the cathode of valve 'B', the discharge of the selected condenser is effected at will either through the sweep circuit or through the trigger.




OSCILLOGRAPH
SWEEP & SHIFT CIRCUIT
FIG. 23

SECTION E. ASSOCIATED EQUIPMENT.

(a) Sweep Trigger and Phase Selector. The mechanical device for regulating the sweep (Figs. 24 and 26) employs the pentode and condensers in the valve time-sweep with their appropriate controls and serves to hold the condensers short-circuited except during the interval of sweep. As shown in Figs. 24 and 26, the device consists essentially of an ebonite commutator fixed to the pump shaft. Two brass segments in the ebonite are so placed that the gaps between them are respectively 1/16 inch less and 1/16 inch greater than the diameter of the carbon brushes working over their faces. The brush-holder carries three contacting brushes (A.B.C.) 5/16 inches diameter of which one - the outer - is mounted on a moveable fitting in a dove-tailed groove cut in the holder. The brush holder can also be moved as a whole relatively to the commutator and clamped in any position. The sweep condenser circuit is completed by connection through either brushes 'A' and 'B' or through 'A' and 'B' coupled, and 'C'.

With the former connection, rotation of brush 'A' relatively to 'B' and 'C' allows of the circuit being short-circuited over a range of pump angle from $180^\circ$ to $360^\circ$ while with the latter a range from $0^\circ$ to $180^\circ$ may be obtained. Since the sweep condensers are only charged during the period of short circuit, the length of the sweep for a given sweep

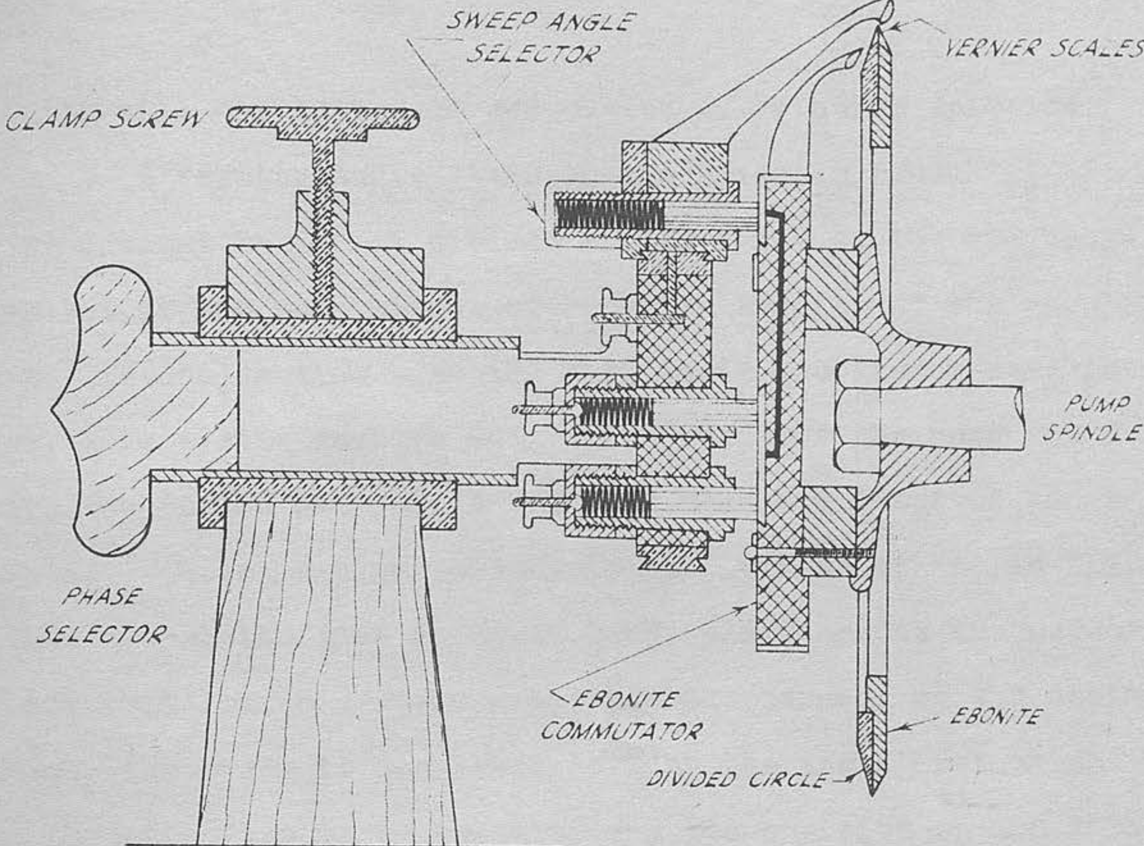



FIG 1

FIG 2

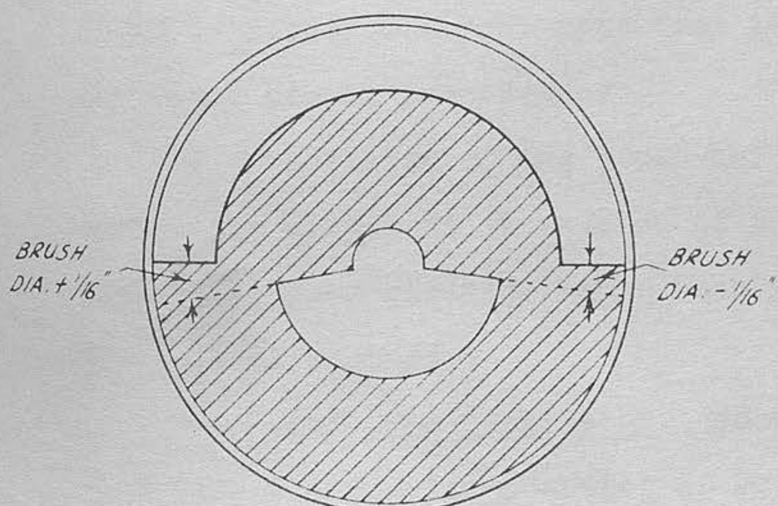
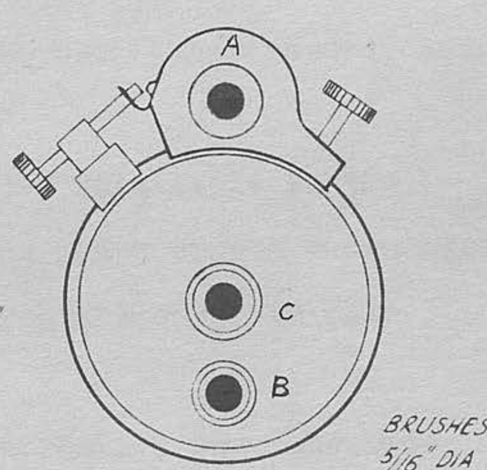



FIG 3




DETAIL OF MECHANICAL TRIGGER

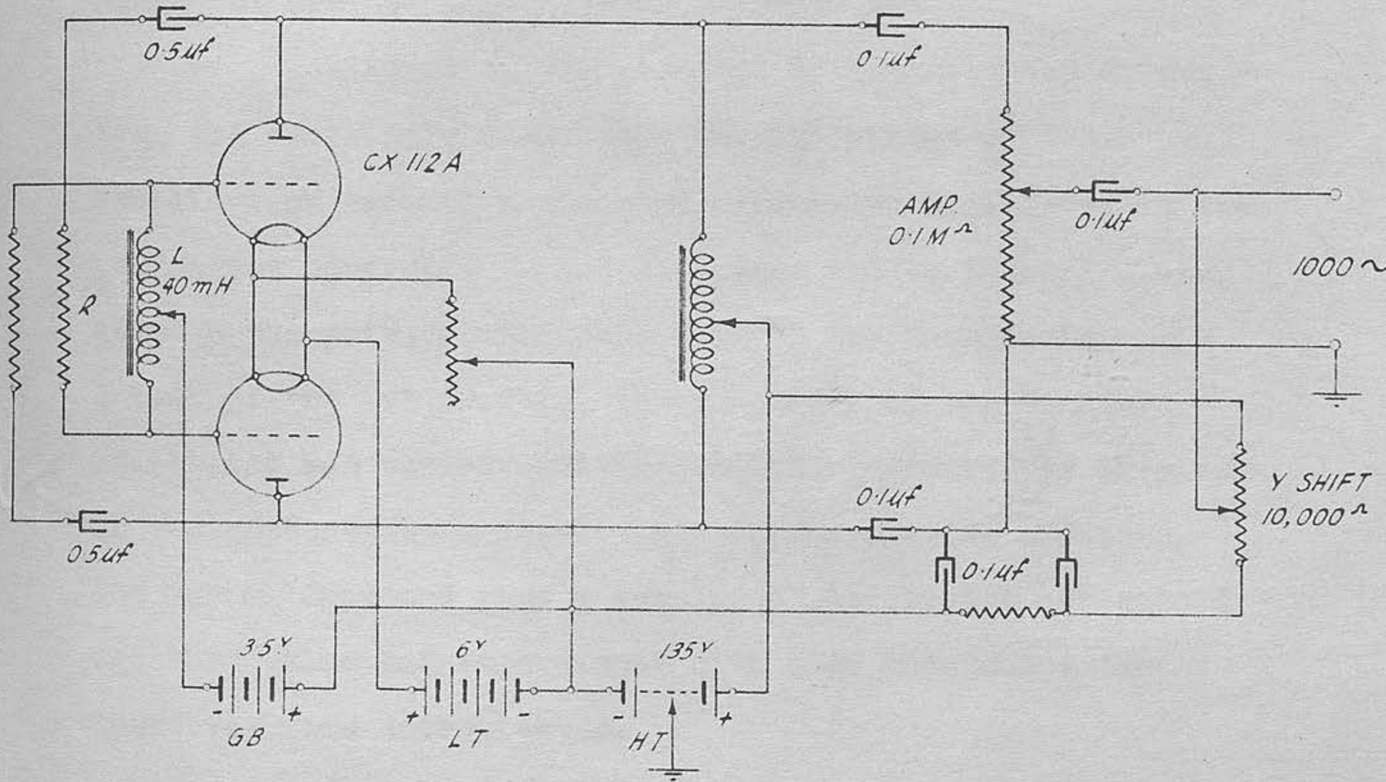
FIG.24.

velocity will correspond to the period of rotation selected.

A Vernier Scale fixed to the moveable brush 'A' moving against an 8-inch divided circle fixed to the end bearing-cover of the pump allows of accurate reading of the angular sweep. In conjunction with the 1000 cycle oscillator described below, this system enables accurate records of the pump speed during the period over which the phenomena are observed to be obtained. Using a high sweep velocity, the brush 'A' is fixed in such a position that no sweep takes place across the screen and its position accurately read with the Vernier on the divided circle. The brush is now moved through the angle over which the pump phenomena are to be observed and its position on the divided circle again noted. The sweep velocity having been adjusted to give the requisite length of diagram on the screen, the time of the sweep in milliseconds corresponding to the selected angle of rotation may be read from the oscillator record, and the speed of the pump deduced.

Rotation of the brush-holder relatively to the commutator allows of alteration of the phase of the sweep with respect to pump rotation, thus enabling the complete pump cycle to be scanned.

(b) The Oscillator. To correlate pump phenomena with time and angular position of rotation, a 1000 cycle time-base was incorporated in the records taken. The oscillator for this purpose (Fig. 25) consists of two battery operated




1000 ~ OSCILLATOR

FIG. 25

Horton resistance stabilised oscillators working in push-pull.

Using resistance coupling, their operation depends on the feed-back of portion of the anode current in correct phase to the tuned grid circuit. The coils are astatically wound but are not connected in reverse mutual.

Calibration was effected by opposing the output from the oscillator coupled to the 'Y' plates of the oscillograph against a standard calibrated oscillator in the Postmaster General's Department connected to the 'X' plates through the private telephone wire to the Engineering School of the University. The frequency of the standard oscillator was varied until a stationary figure was obtained on the Cathode-Ray Screen. When this condition obtained, the P.M.G. Standard gave a reading of 960 cycles per second and this value has accordingly been used throughout the tests for time determinates.

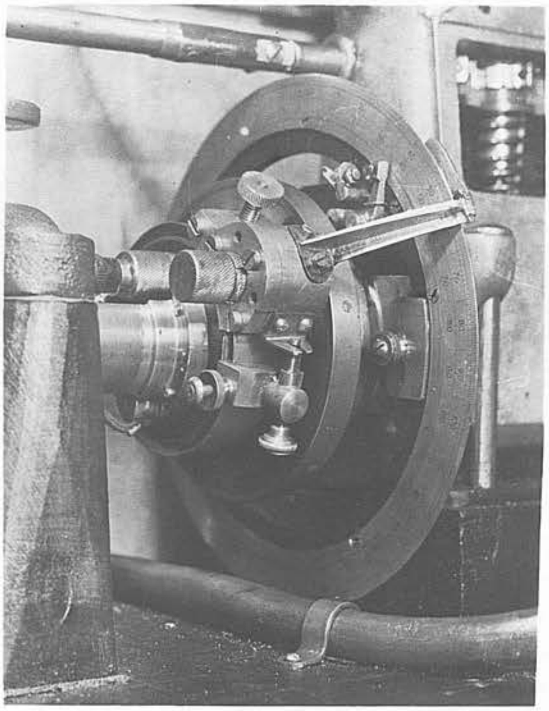


FIG. 26.

MECHANICAL TRIGGER.

and

PHASE SELECTOR.

(c) CRYSTAL OUTPUT AMPLIFIERS.

(1) Power Supply. The power-supply unit for the amplifiers (Fig. 27) embodies a neon stabilised rectifier to prevent mains variation and to prevent the drain of one amplifier affecting the out-put of the other.

A full wave rectifying valve across the 385 volt. tapping of the secondary of the mains transformer supplies some 300 volts d.c. to the filter consisting of two 8 micro-farad electrolytic condensers and a 40 Henry choke. Through a 30,000 ohm voltage divider across the busbars some 90 volts are tapped and stabilised with an 874 neon stabiliser which will pass from 10 to 40 milli-amperes with but slight variation in voltage. The voltage was set by adjusting the voltage divider tapping to pass 20 milli-amperes through the neon tube. This supplies the anode voltage and a tapping at 60 volts the screen voltage. A closed circuit jack connected in the anode supply for each valve enables the plate current to be read, thus ensuring operation of the amplifiers within the working range of the Eg/Ia characteristic curve (Fig. 28).

(2) AMPLIFIERS.

The amplifiers (Figs. 27 and 29) totally enclosed in iron shielding-boxes are placed as close as practicable to the crystals to which they are connected by leads of very low capacity. The bias batteries for each amplifier are contained within the shielding boxes and connected through potentiometers to permit of relative biassing of the pressure

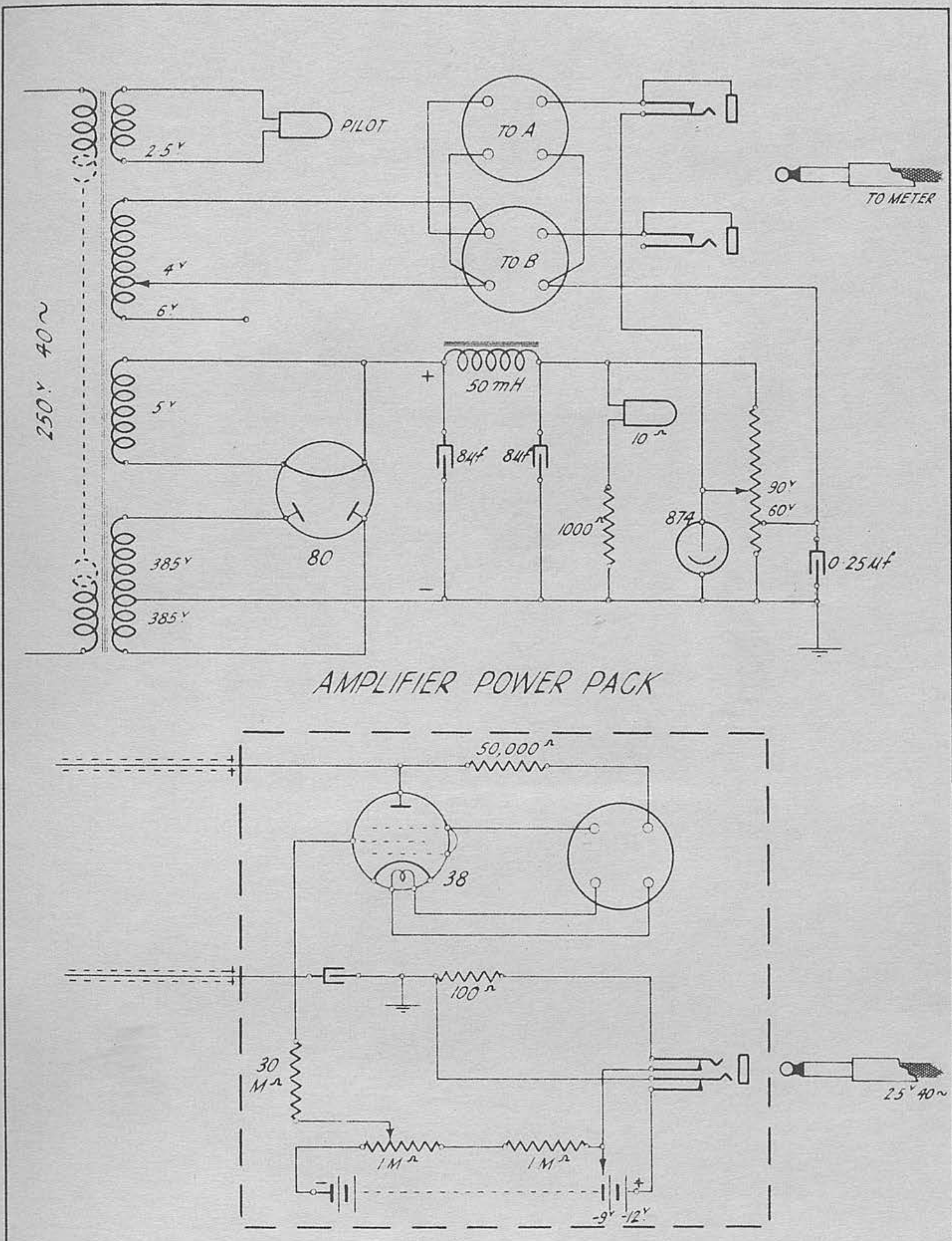

PILOT
2·5 V
TO A
TO B
TO METER
4 V
6 V
250 V 40~
5 V
385 V
385 V
80
50 mH
8μf
8μf
10 Ω
1000 Ω
874
90 V
60 V
0·25 μf
AMPLIFIER POWER PACK
50,000 Ω
38
100 Ω
30 MΩ
1 MΩ
1 MΩ
-9 V
-12 V
25 V 40~
AMPLIFIER

FIG. 27

diagrams. Grid bias is obtained through use of a grid leak and in order to maintain the time-constant of the input circuit, a valve of 30 megohms was selected. As the largest leaks available were of the order of 5 megohms, a special type was made consisting of platinum wire fused in a bent 4 m.m. glass tube filled with commercial Amyl Alcohol (Fig.29)

The wiring of the closed circuit jack is such that when 40 cycle current of about 2 volts potential is plugged in, the grid bias is automatically set to the mid point of the Eg/Ia characteristic curve, so giving a completely amplified 40 cycle diagram on the screen. Us-ing a standardised input voltage, a negative of the oscillogram replacing the ground glass screen of the camera, serves as a basis for periodical checking of the overall amplification.

(d) NEEDLE-LIFT INDICATOR (Figs.12 and 32).

While the method adopted for delineating the needle lift does not lend itself to accurate scaling, it possesses the distinct advantage of not adding to the weight of the moving parts whereby inertia factors of a relatively high order are introduced, and may therefore be expected to give a faithful reproduction of the needle movement. In the Bosch injector, the spring load is transferred to the needle through an extension spindle bearing on the needle. The head of the spindle losated within the loading spring follows

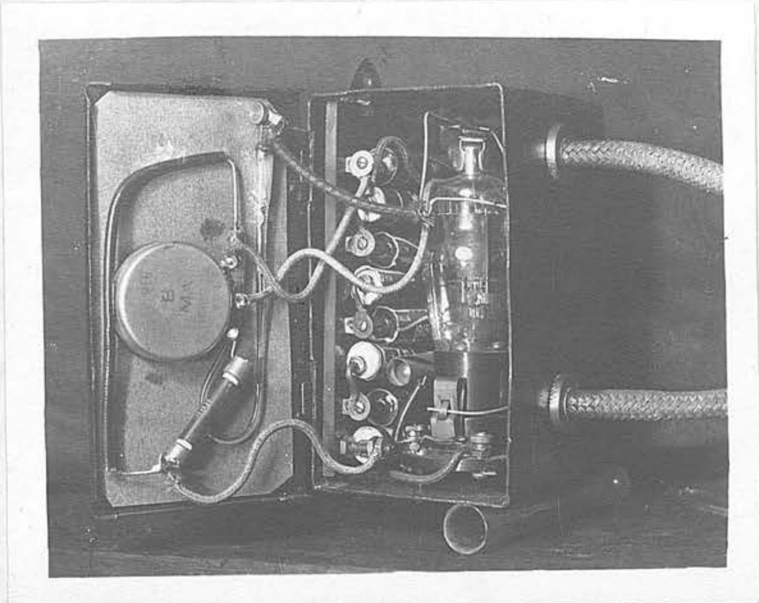


FIG. 29.
AMPLIFIER.

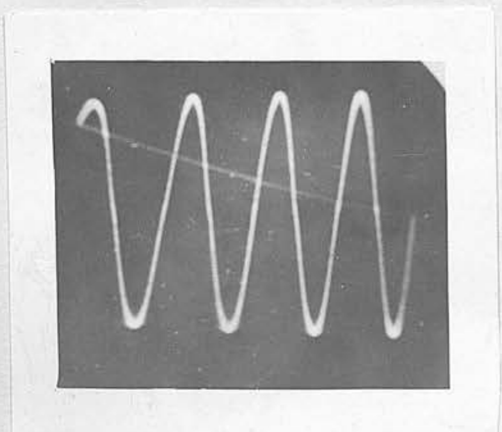


FIG. 30.
40C/S.

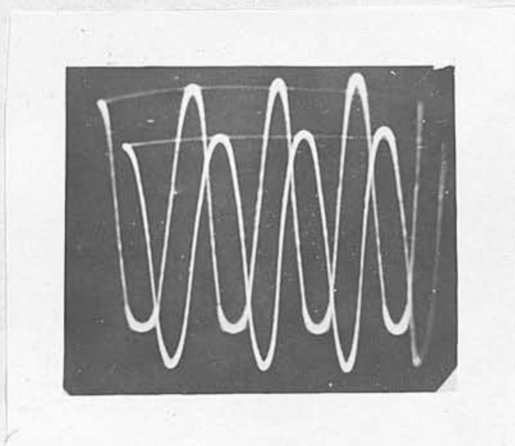


FIG. 31.
40C/S. OUTPUT.
from AMPLIFIER.

needle movements in a reasonably faithful manner and the principle employed depends upon the variation of electrical resistance when current flows between the moving spindle head and a fixed insulated electrode immersed in a conducting fluid filling the body of the nozzle.

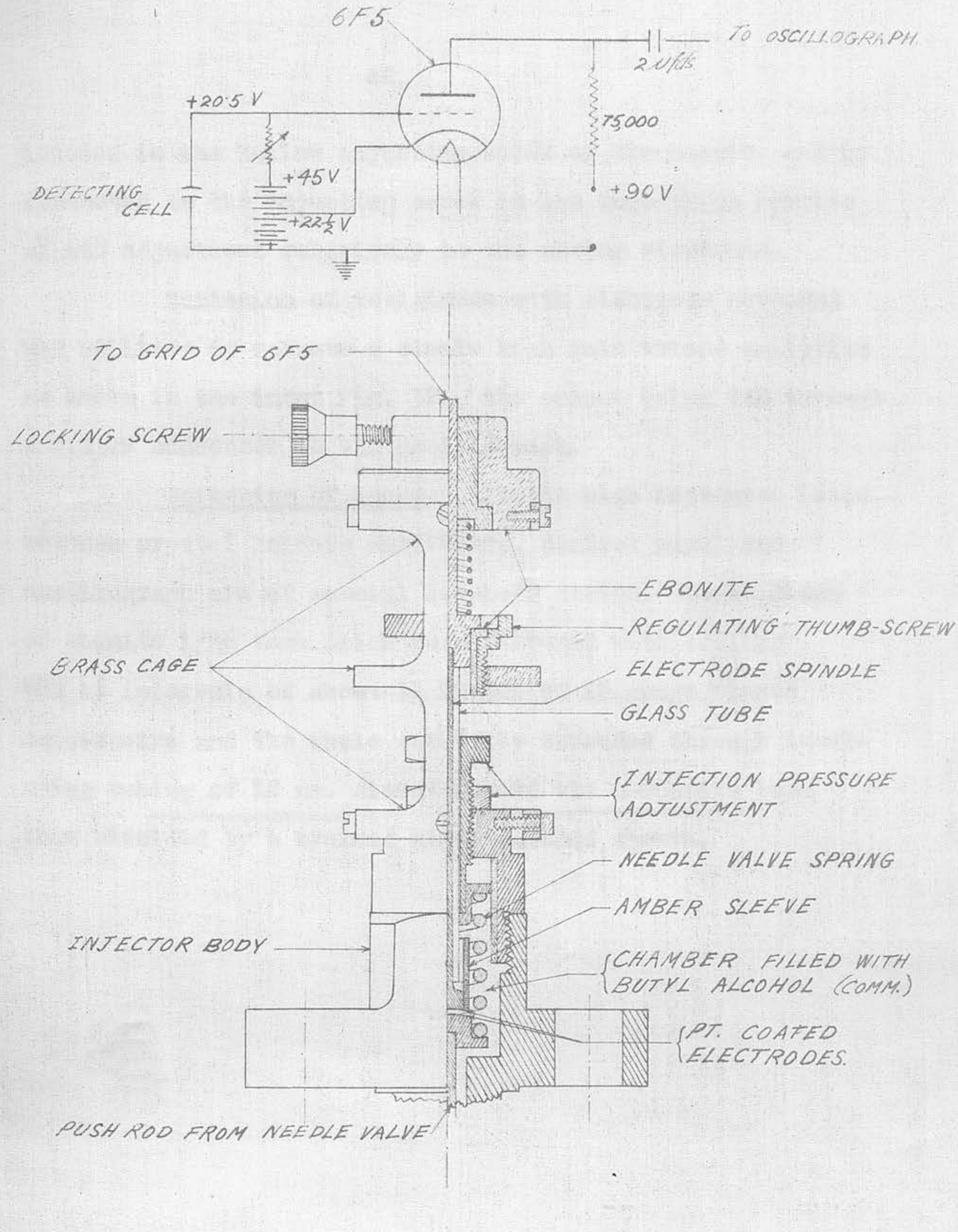
Some considerable time was spent in determining a suitable non-corrosive liquid of sufficiently low conductivity and commercial Butyl Alcohol was finally selected as being the most satisfactory medium.

Full needle movement is of the order of ½ millimetre and a liquid with low conductivity is essential in order that a small variation in the electrode gap may produce relatively large variations in resistance with a reasonably small applied voltage. The cap of the spindle forming the lower electrode was ground to a radius corresponding to the distance between the top of the cap and the point of contact of spindle and needle valve and the fixed electrode ground concave to a slightly larger radius.

This procedure serves to maintain a constant gap between the electrode faces irrespective of oscillatory movement of the spindle about the point of contact with the needle valve.

Both electrodes were faced with platinum foil.

The fixed electrode is carried on a steel wire 1/16 inch diameter which passes through a 2mm. glass tube




DETAILS OF
NEEDLE VALVE LIFT DETECTOR

FIG. 32.

located in the hollow adjusting screw of the nozzle, and is connected to the adjusting screw in the cage which permits of its adjustment relatively to the moving electrode.

Variation of resistance with electrode movement was utilised to actuate a simple high gain triode amplifier as shown in the inset Fig. 32., the output being fed through a 0.2 $\mu$F condenser to the oscillograph.

Screening of Leads. All the high impedance leads between crystal holders amplifiers, control panel and oscillograph are of special low-loss design. Small discs of ebonite 1/16 inch thick were fastened with sealing wax at intervals of about 1¼ inches to 18 gauge tinned copper wire and the whole carefully threaded through insulating tubing of 12 mm. diameter, and the composite lead then shielded by a braided steel earthed sheath.

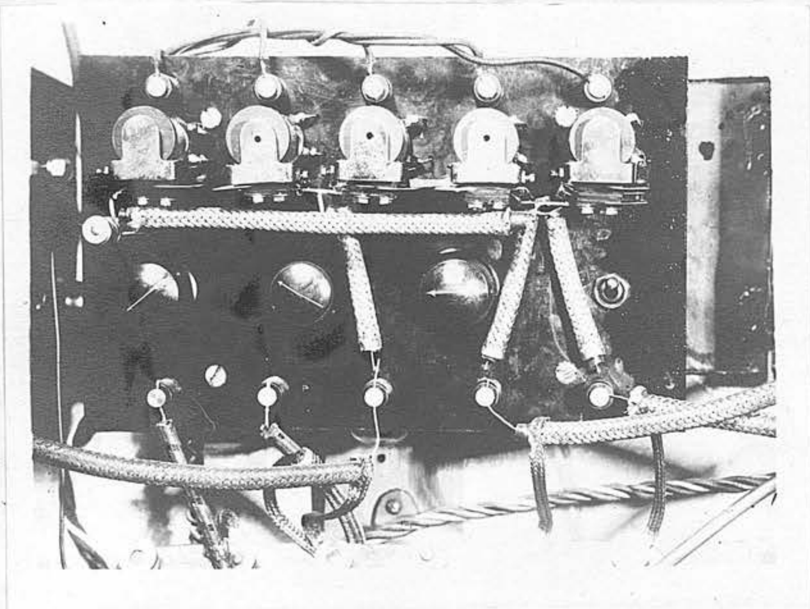


FIG. 33.

RELAY PANEL.

PHOTOGRAPHIC RECORDING.

In developing the photographic technique, it was considered desirable to record the observed phenomena of pressure and needle lift under any set conditions of speed and load in as short a time as possible, and if possible, on the same photographic plate to facilitate ready correlation and comparison.

The system adopted employs d.c. operated relays operated through a commutator in preference to manipulation of camera shutter, and the three records consisting of two pressure diagrams and needle lift diagram are photographed within the time required for three revolutions of the pump.

The relay panel (Fig. 33) is placed close to the pressure recording units to give the shortest possible leads. Five relays of the post-office telephone type are employed, and, reading from right to left (Fig. 33) are biassing, pressure (pump) pressure (nozzle) needle-lift indicator and oscillator. The controls reading from left to right are relay selector switch, relay driving-voltage control, bias potentiometer and bias switch. At the bottom of the panel are the work terminals and at the top, the driving-voltage terminals from the commutator.

Power for driving the relays is taken from the

D.C. generator and fed through the 500 ohm. voltage divider (Fig. 34). One end of this divider is taken through a 50 ohm rheostat to the centre point of a six-point switch which enables either relay to be closed at will, thus enabling individual inspection of the diagrams, or when connected to position 5, places the relays under control of the commutator and the diagrams appear on the screen in rotation.

The commutator (Fig. 35) consisting of a disc of ebonite with a 90° brass segment inlaid, is driven from the pump through a 4:1 gear reduction. Four light spring brushes connected to corresponding relays in the relay panel (Fig. 34) make contact with the brass segment throughout one revolution of the pump. The circuit of the energising current for the relays is completed through the brushes, thence to the earthed frame of the commutator and through a flexible arm normally earthed to position 6 of the selector switch, thus energising each relay in turn during one revolution of the pump. When the flexible arm is insulated from earth, the bias relay comes into operation, and the common work-lead from the other relays to the work terminal of the oscillograph receives a bias of sufficient magnitude to drive the moving spot clear of the screen.

This feature is utilised during photography in the following manner. On the commutator spindle, there is fixed a composite worm of $\frac{1}{4}$ inch pitch. The first complete thread is of ebonite followed by a complete thread of brass and the worm is so fixed on the commutator spindle that during rotation, the junction coincides with the contacting of the commutator brush for operating the pump pressure relay with the $90^{\circ}$ segment on the commutator. When the flexible arm is released from its normal earthed position, it comes in contact with the ebonite thread and works slowly across the worm. Being insulated from earth, the bias relay operates, the spot is biassed clear of the screen and the photographic film which has now been exposed to the screen will be unaffected. When, however, the moving arm contacts with the brass thread of the work, the bias is removed and the pressure near the pump is recorded on the exposed film to be followed by that at the nozzle, the needle lift and the 1000 cycle time base as the corresponding relays are energised.

On the completion of the revolution of the commutator (corresponding to four revolutions of the pump) the four diagrams will have registered on the exposed film.

At this point, the moving arm springs clear of the work, the bias relay operates and the spot completely biassed off the screen thus preventing fogging of the film.

The most suitable photographic medium for recording the transients was found to be X-Ray film, which is specially prepared for operation with intensifying screens.

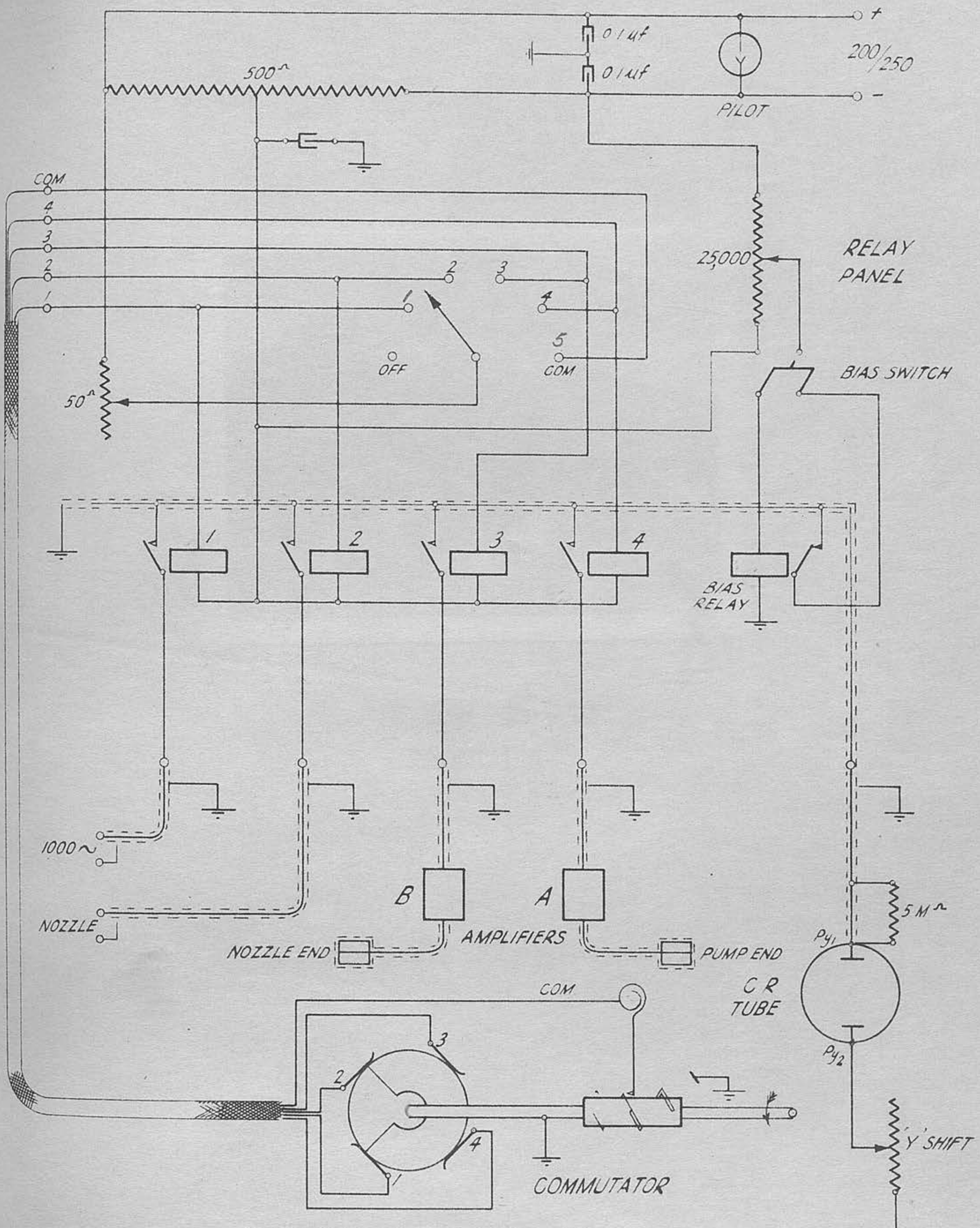



DIAGRAM OF CONTROL PANEL

FIG. 34

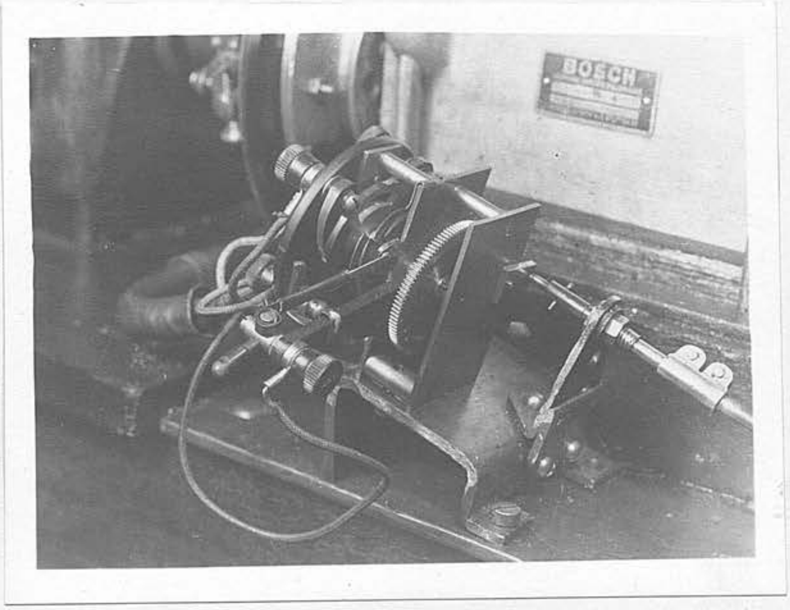



FIG. 35.

COMMUTATOR.

## PART IV.

## EXPERIMENTAL RESULTS.

### Section I

(i) Nature of Investigations.

(ii) Data relating to Injection and Pressure-measuring Components.

(iii) Calibration.

## PART IV. EXPERIMENTAL RESULTS.

### SECTION I.

(i) Nature of the Investigations : The success of an injection system may be gauged by the nature of the quantity, time and pressure factors associated with the injection of the oil throughout the speed and load range of the engine. These factors, termed the characteristics of a particular injection system may be designed or accidental. The latter may occur under certain conditions of working unforeseen in the original design of the injection system and may be detrimental or otherwise to engine performance. The experimental investigations to be described deal primarily with the influence of certain variables upon the injection characteristics of a Bosch Fuel Injection System.

The elements common to injection systems comprising the pump and the injection nozzle may be regarded as the major variables in a general survey of the problem, but for a particular system, they become the fixed components and such variations in discharge characteristics as may occur throughout their working range of speed and load are inherent in their design. These may be influenced however, by the dimensions of the connecting pipe between pump and nozzle, the nature of the fluid being pumped and the pressure of the engine cylinder acting on the nozzle. The influence of the two latter variables on the discharge characteristics has not been considered in the present research. While the back pressure must undoubtedly influence the discharge, it was considered that the effect would not be of such an order as to appreciably affect the results. The effect of differences in viscosity and bulk modulus between various fuel oils on the injection process has been investigated by several workers including Le Mesurier and Stansfield[11] and Davis and Giffen[4]. The latter workers have shown that the modifications to the injection characteristic resulting from the use of fuels of widely varying viscosities at normal temperatures are negligible except in the case of very long pipes.

This result might have been anticipated from a consideration of the bulk moduli and densities of the fuels used, since these do not differ greatly and the Pressure-Velocity relationship is governed by the equation :

$\frac{\text{Change of Pressure}}{\text{Change of Velocity}} = \sqrt{\frac{K\rho}{g}}$ . The variation in the discharge characteristics when thefuel is raised to temperatures likely to obtain in practical operation is probably a much more serious matter than variation with viscosity, and is a subject which the author hopes to investigate later.

Exclusion of the back-pressure and fuel variables confines the practical variables to those of the interconnecting pipes. In the present work, three steel pipes of standard commercial quality 2mm. nominal bore and 2ft., 4ft., and 8ft. long respectively have been chosen to represent the connection variable, and the discharge characteristics have been studied, through as large a working range of speed and load as practical application would seem to warrant, and the recording apparatus permit. In this regard, it is to be regretted that the maximum pump speed is not higher, but a limitation was set by the inability of the valve lift recorder to give accurate records at speeds greatly in excess of 900 r.p.m. Beyond this speed, vibrations were apt to induce instability of operation and consistency of results could not be depended upon.

For each length of connecting pipe, the pump speed was varied in five steps from about 200 rp.m. to 900 r.p.m. and at each speed step, the load was varied as indicated in Fig. 36.

LOAD VARIATION.

Speed. (Nominal.).

200 330 450 700 900

CHARGE (Fraction of Full Charge)

$\frac{1}{4}$ $\frac{1}{2}$ $\frac{3}{4}$

Nozzle Setting (Atmosphere) | Nozzle Setting (Atmosphere). | Nozzle Setting. (Atmosphere).

100 150 200 100 150 200 100 150 200

FIG. 36.

For each speed and load setting, the pressure phenomena at pump and nozzle, the valve lift and the time base were photographically recorded in the manner previously described, the gun voltage being kept constant at 600 throughout the whole series of recordings. The fuel discharge in cubic centimetres per stroke was measured for each setting by taking accurate speed and time observations while about 50 cubic centimetres of the fuel were drained from the graduated burette. The exact quantity drained was determined after sufficient time had elapsed to allow of the gravitation of the fuel adhering to the sides of the burette. At low rates of withdrawal, this quantity was negligibly small but became increasingly important as the pump speed and throttle opening increased.

(ii) Data relating to Injection and Pressure-Measuring Components:

Fuel Pump: Standard Bosch, Type P.E. with unloading valve. (Fig. 37(a) ). Plunger diameter: 7mm. Plunger Stroke (Fig. 38).

Nozzle: Bosch Spring - loaded Differential Needle Atomiser - Hole type. Style No. D.L.O. S288-1OP. (Fig. 37(b)).

Pipes: Standard Commercial Steel Pipe 2mm. nominal bore.

Dimensions: (1) Length : 2 feet. Capacity, 1.61 cubic centimetres. Mean Bore (Calculated from length-capacity) 1.83 mm.

(2) Length : 4 feet. Capacity 4.45 cubic centimetres. Mean Bore 2.16 mm.

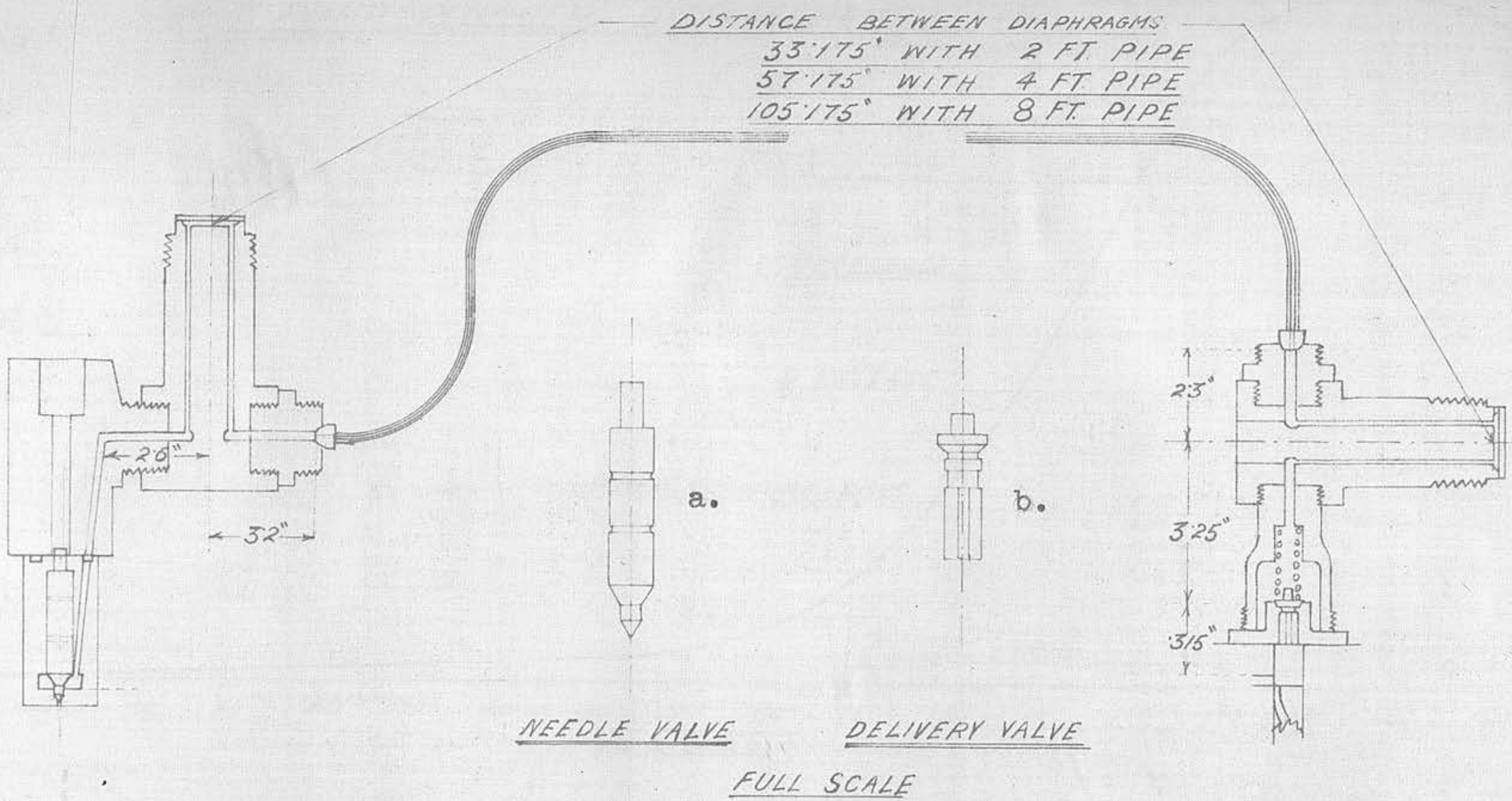
(3) Length : 8 feet. Capacity, 6.61 cubic centimetres. Mean Bore 1.86 mm.

Capacities: The capacities of the components shown in Fig. 37 were determined by filling with mercury. The figures given are the mean of four very close determinations.

Pump Plunger Stroke and Velocity Diagram (Fig.38) The plunger stroke on an angle base was obtained by direct reading with a measuring microscope reading to 1/1000 mm. The velocity diagram giving velocities in millimeters per degree was obtained by graphically differentiating the lift curve.

Distance between Pressure-Measuring Points (Fig.37) The actual distances between the diaphragms of the pressure-indicators for the nominal pipe lengths used are:

| | | | | |
|---|---|---|---|---|
| (a) | Two Feet Pipe | : | 33.175 | inches. |
| (b) | Four Feet Pipe | : | 57.175 | " . |
| (c) | Eight Feet Pipe | : | 105.175 | " . |




Capacities:
(a) Pump Barrel from release port to Delivery Valve seat. 0.468 c.c's.
(b) Between Delivery Valve to inlet to first Crystal Holder. 1.4c.c's.
(c) Between inlet to first Crystal Holder and inlet to fuel pipe. 0.6176 c.c's.
(d) Fuel Pipes:

| | | |
|---|---|---|
| 2 feet | : | 1.61 c.c's. |
| 4 " | : | 4.45 c.c's. |
| 8 " | : | 6.61 c.c's. |

(e) Between pipe inlet to No. 2 Crystal Holder and inlet to Nozzle Holder. 0.636 c.c's.
(f) Between inlet to nozzle holder to nozzle face. 0.4c.c's.
(g) Nozzle face to jet plate. 0.39 c.c's.

FIG. 37.

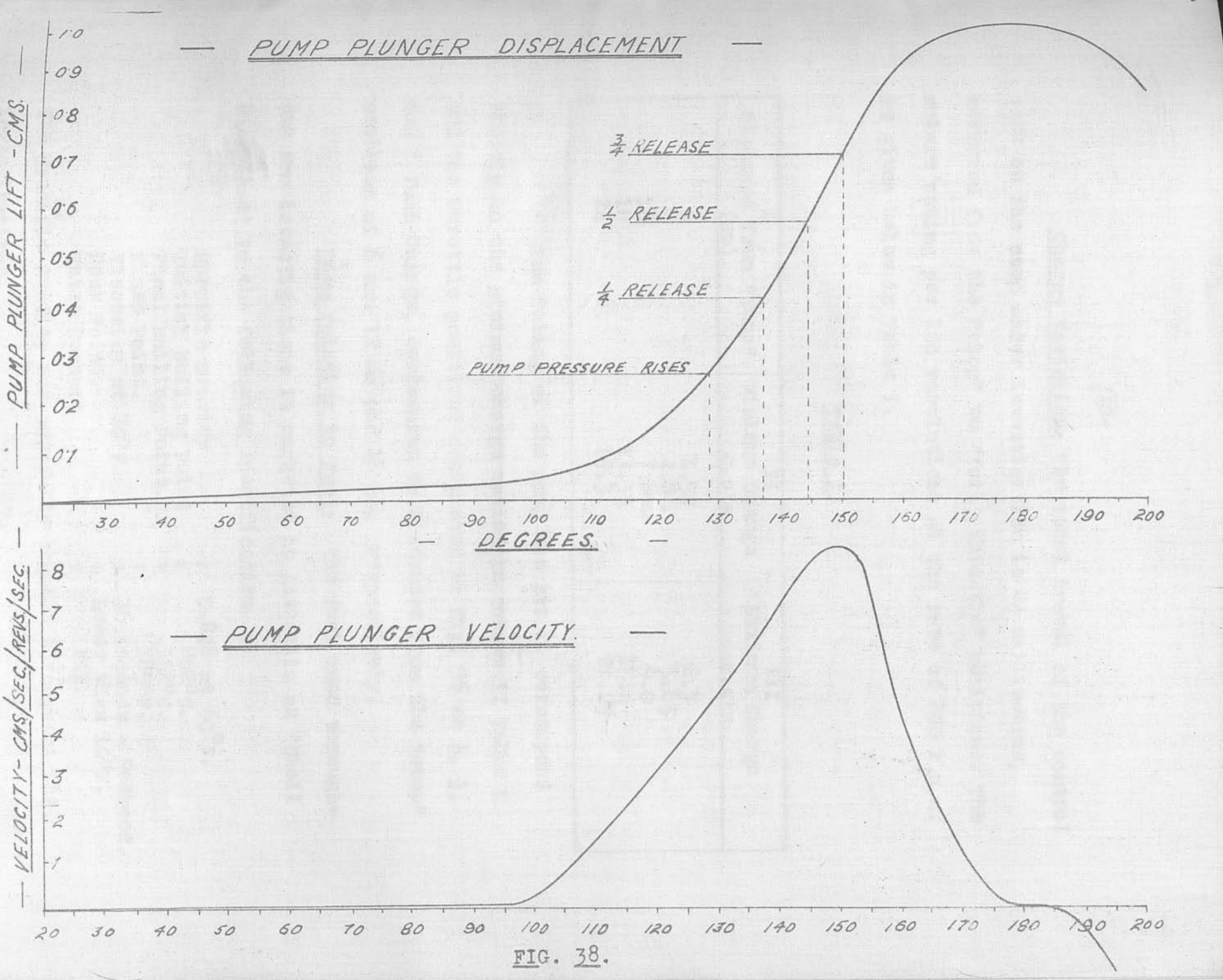

PUMP PLUNGER DISPLACEMENT
PUMP PLUNGER LIFT - CMS.
1·0
0·9
0·8
0·7
0·6
0·5
0·4
0·3
0·2
0·1
3/4 RELEASE
1/2 RELEASE
1/4 RELEASE
PUMP PRESSURE RISES
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
DEGREES.
PUMP PLUNGER VELOCITY
VELOCITY - CMS/SEC./REVS./SEC.
8
7
6
5
4
3
2
1
20

FIG. 38.

<u>Charge Variation</u>: The total travel of the control rack on the pump under investigation is 24 millimetres, measured from the "stop" to "full throttle" positions. The makers'rating per 100 revolutions at the rate of 200 r.p.m. is given below in Table I.

Table I.

| I<br>Distance from "Stop".<br>(mm). | II<br>Minimum Charge<br>c.c's | III<br>Maximum Charge<br>c.c's |
|---|---|---|
| 6 | 1.59 | 2.7 |
| 9 | 4.8 | 5.85 |
| 12 | 7.6 | 9.0 |
| 18 | 13.35 | 15.4 |
| 24 | 19.3 | 21.25 |

The rating of the pump was set to correspond closely to the minimum charge rating in column II Table I and the throttle positions designated in Fig. 36 as ¼, ½, and ¾ Full Charge, correspond to distances from the "stop" position of 6 mm. 12 mm and 18 mm. respectively.

<u>Data relating to Fuel</u>: The fuel used throughout the investigations is marketed in Australia as "Shell Diesoline" to the following specification.

| | | |
|---|---|---|
| Specific Gravity | : | 0.848 at 60°F. |
| Initial Boiling Point | : | 202°C. |
| Final Boiling Point. | : | 352°C. |
| Flash Point. | : | 185°F. |
| Viscosity at 70°F. | : | 36 seconds - Redwood. |
| Pour Point. | : | Lower than 10°F. |
| Water Content. | : | Nil. |

The Specific Gravity was carefully checked throughout the investigations.

(iii). Calibration. Static Calibration of the apparatus is obviously impracticable on account of the small time-constant and consequent leakage of charge during the loading process . Dynamic Calibration presents many difficulties. Several methods were tried including the use of calibrated rupture discs. A small disc holder was made in which a thin metallic disc could be clamped between ground joints in a manner similar to thediaphragm mounting in the crystal holders leaving ¼ square inch of disc exposed to the oil pressure, when the holder was connected to the outlet of the crystal holder as shown in Fig. 39. The pump was turned by hand and the pressure raised to the point of rupture of the disc. The sudden release of pressure to atmospheric caused by the rupture gave a negative deflection to the spot focussed on the screen, and the length of the trace gave a measure of the difference in pressure before and after rupture. In spite of every care exercised in the selection of the rupture disc material as regards uniformity in thickness, results were not consistent, varying by as much as fifteen per cent. The method finally adopted is simple in application, and gives consistent results.

The principle of the method depends upon an observable phenomenon associated with the actual injection process. When the needle of the atomiser lifts as the

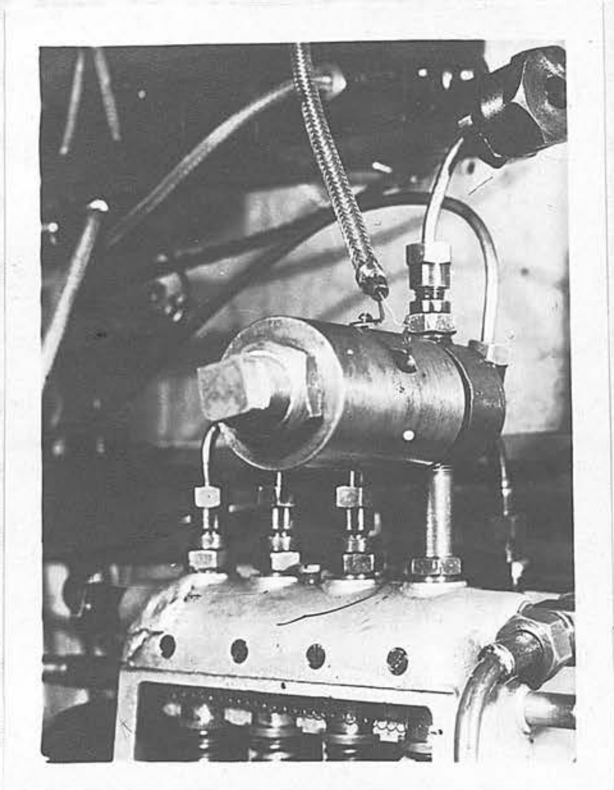


FIG. 39.

RUPTURE DISC MOUNTING.

result of a pressure impulse received from the pump, there is a sudden brief drop in pressure owing to the enlargement of the pressure space by the capacity of the jet plate and the needle movement. The pressure before the drop occurs will be greater than the set opening pressure of the needle by an amount depending on the inertia of the needle, the needle extension and loading spring, and the reasonable assumption is made that for set pump speed and charge conditions, this inertia load remains sensibly constant throughout the needle-loading range.

Under these conditions, the height of the recorded pressure diagram when the drop occurs at needle lift will represent the set opening pressure of the needle, plus a constant, and by plotting a series of records at various needle loadings, this constant is determinable thus enabling a pressure scale for the records to be made.

By the simple process of taking a photographic record under the speed and charge conditions on which the pressure scale was based, it is possible at any time during the investigations to check the calibration.

At the commencement of the investigations, both crystal holders were calibrated to a maximum pressure of three hundred atmospheres in steps of fifty atmospheres and check calibrations were made at frequent intervals during

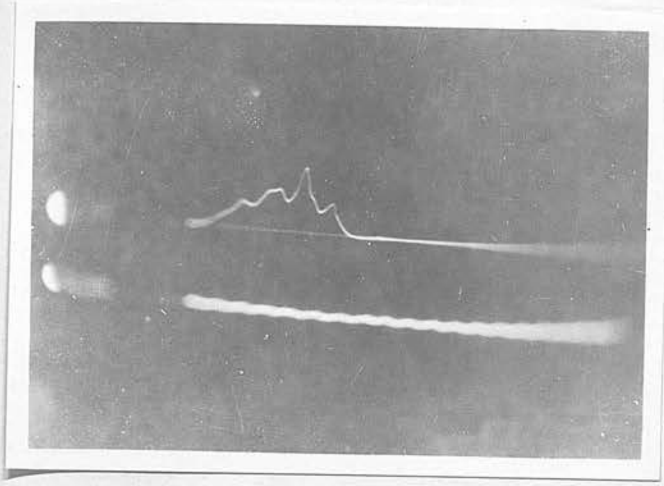


FIG. 40.
50KG per $CM^2$.

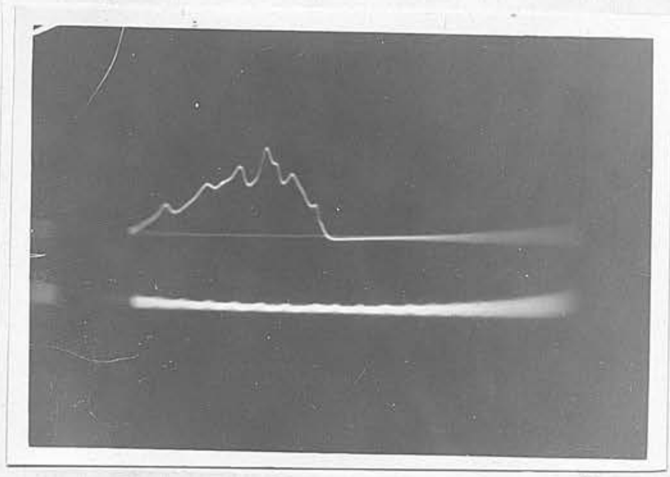


FIG. 41.
100KG per $CM^2$.

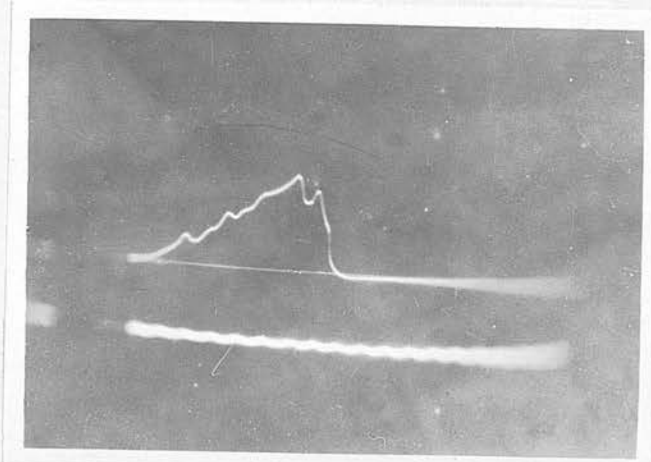


FIG. 42.
150KG per $CM^2$.

FIG. 40-42.

CALIBRATION RECORDS.

No.1. CRYSTAL HOLDER.

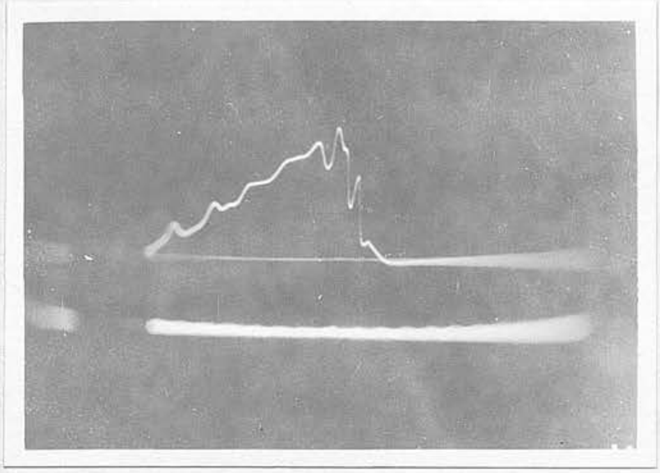


FIG. 44.
200KG. per $CM^2$.

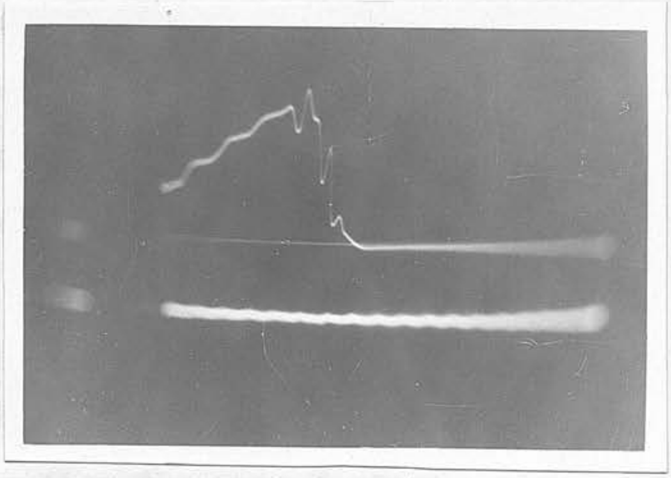


FIG. 45.
250KG. per $CM^2$.

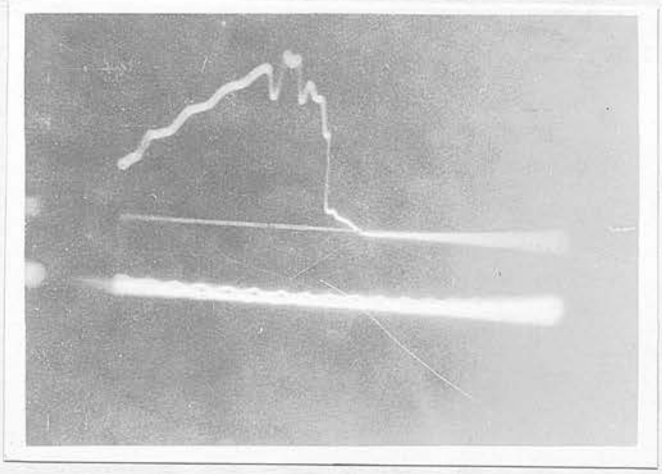


FIG. 46.
300KG per $CM^2$.

FIGS. 44 - 46.

CALIBRATION RECORDS.

No. 1 CRYSTAL HOLDER.

the progress of the work. Slight variations were noted from time to time, but were of such a small order as not to warrant departure from the very desirable feature of a uniform pressure scale. A typical set of calibration records are shown in Figs. 40-46, and the pressure scale for both crystal sets in Fig. 47.

The linear response of the crystals to pressure variation is well brought out in the scales and the similarity of response, taking into consideration the many factors which influence the direct output is rather remarkable.

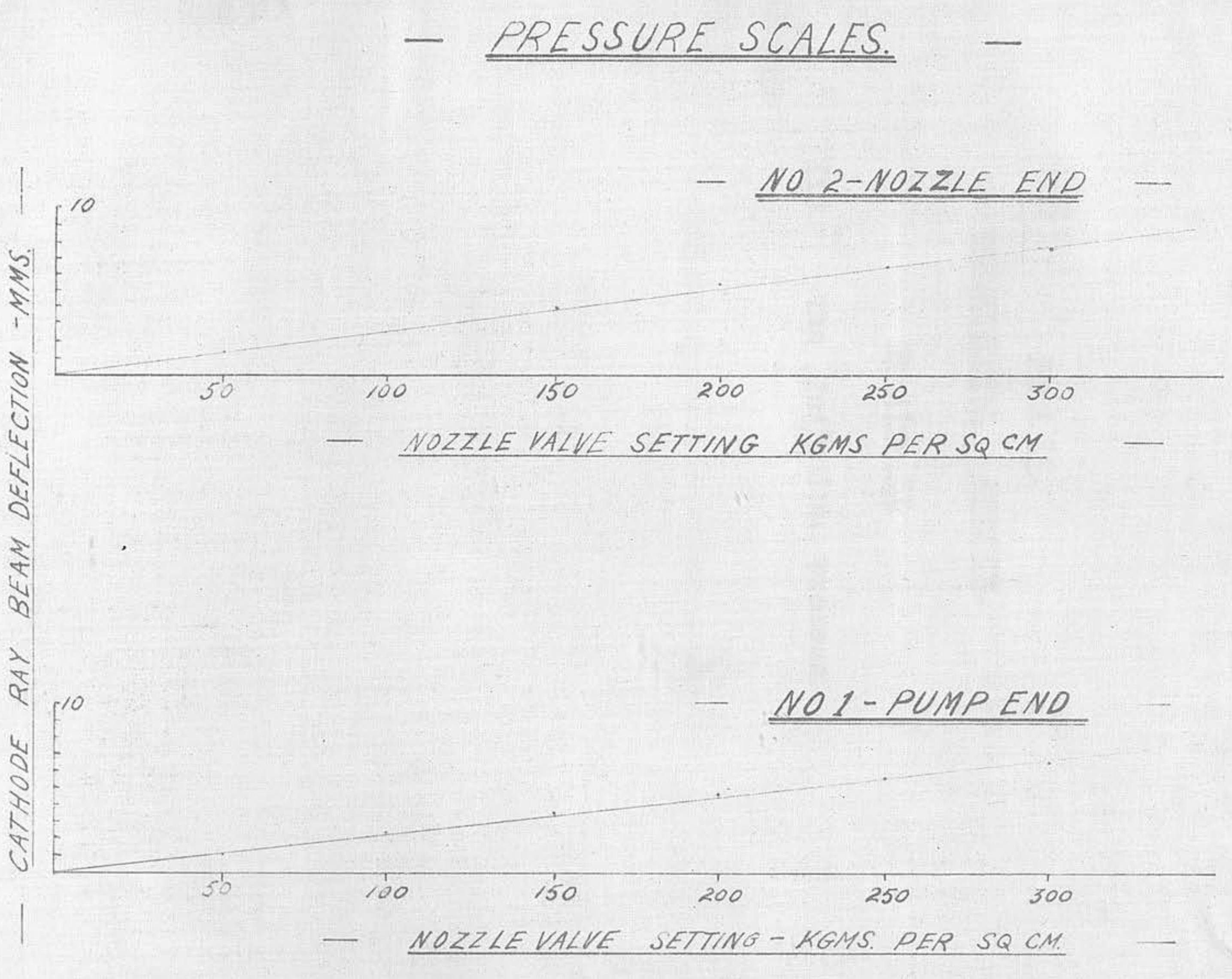

PRESSURE SCALES.
NO 2-NOZZLE END
10
50
100
150
200
250
300
NOZZLE VALVE SETTING KGMS PER SQ CM
NO 1 - PUMP END
10
50
100
150
200
250
300
NOZZLE VALVE SETTING - KGMS. PER SQ CM.
CATHODE RAY BEAM DEFLECTION - MMS.

FIG. 47.

# PART IV.

# EXPERIMENTAL RESULTS.

## Section II

### (i) The Injection Process.

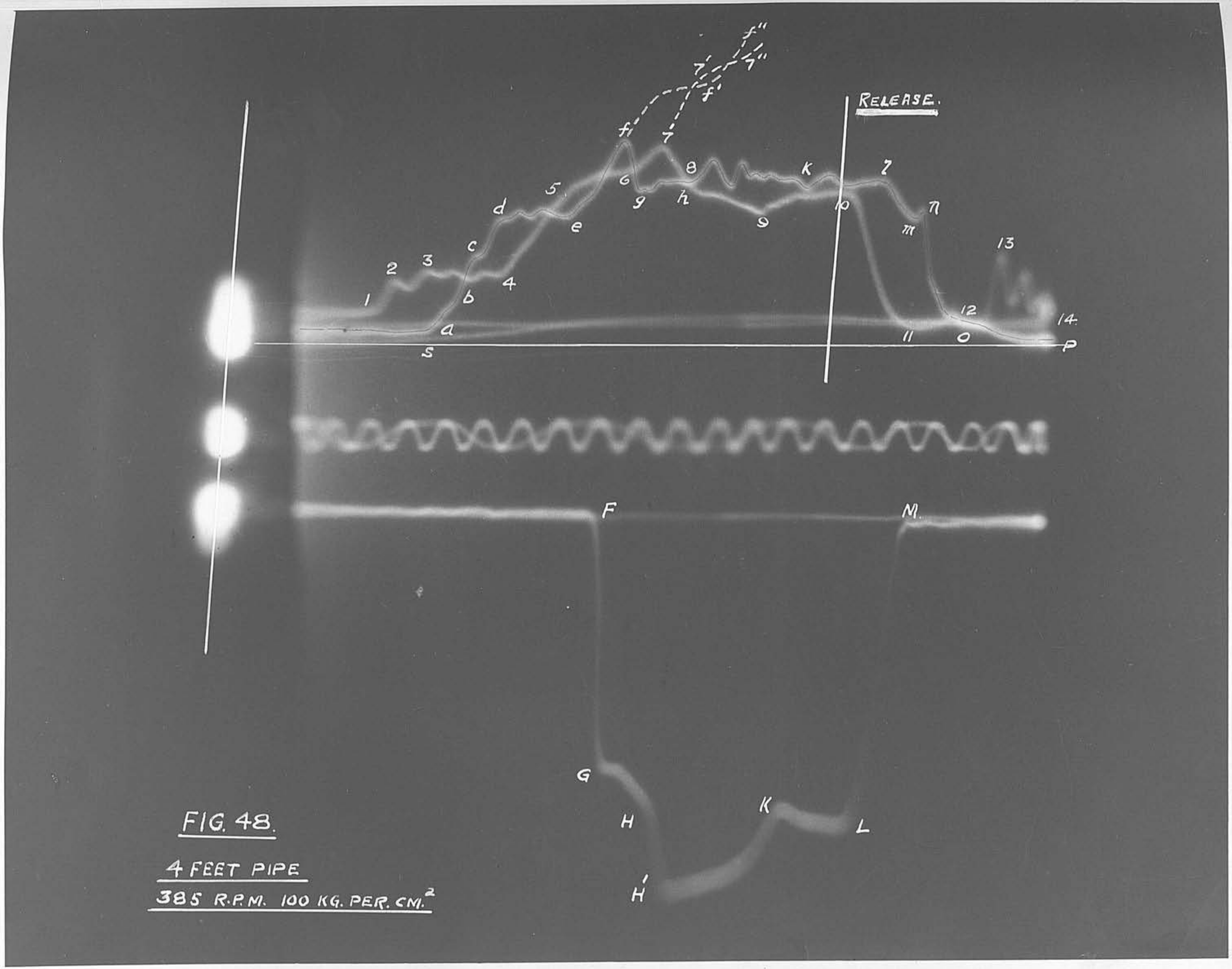



FIG. 48.

4 FEET PIPE

385 R.P.M. 100 KG. PER. CM.$^2$

## EXPERIMENTAL RESULTS.

### SECTION II.

The Injection Process. To facilitate an explanation of the fundamentals of the injection process in the Bosch system as observable from a study of the experimental records, a typical pressure and valve-lift diagram has been enlarged (Fig. 48) and such features as are common to the process as a whole will be referred to this particular record. Where necessary, special features will receive specific reference.

For greater distinction, the pressure diagram taken at the nozzle has been tinged in red. In this regard, it is to be regretted that the conditions of photography did not permit of a biassing apart of the two pressure diagrams as on first approach to a study of the phenomena some difficulty in distinguishing the curves is experienced. This difficulty is however overcome after a short application especially if observations are made under slight magnification, and, in the correlation of pump and nozzle pressure phenomena, the superimposition of the curves is of distinct advantage, more especially since they have a common scale of pressure. The records in Fig. 48 represent the pressure and valve lift diagrams for the four-feet fuel pipe at $\frac{3}{4}$ full charge, 330 r.p.m., and needle valve setting of 100 atmospheres and the phenomena are recorded over a period of $60^\circ$

of pump rotation.

At point (1) on the pump pressure diagram corresponding to time (t) = 0; the first pressure rise is recorded at the first measuring point situated near the pump delivery valve. This point is not that at which the pump plunger covers the suction port, as there exists in the fuel line a residual pressure indicated by s.a. On first covering the suction port, the plunger must therefore move through a certain distance (varying with the residual pressure in the fuel line) until the fuel in the pump barrel is compressed to a pressure exceeding that in the line, thus enabling the discharge valve to lift and free flow to take place. The magnitude of the initial compression thrust 1-2-3 entering the pipe is determined by the velocity of the pump plunger (and hence the resultant velocity in the fuel line) when these pressure conditions are complied with, increasing with increasing plunger velocity . In the particular system investigated, this initial pressure thrust in the line is transmitted from the pump through the unloading valve (Fig. 37(a) ) which, embodies a solid piston finely ground to fit the cylinder in the delivery-valve housing, and thus serves to isolate pump chamber from fuel line.

The compression thrust (1-2-3) is now propagated as a pressure wave through the fuel (as successive increments acquire momentum) with a theoretical velocity equal to that of sound in the same medium and of value $v = \sqrt{\frac{Kg}{\rho}}$ where K is the bulk modulus and $\rho$ the density of the fuel.

Shortly after reaching the second measure point at 'a' , the pressure impulse reaches the needle valve. At this instant, had the motion of the plunger been uniform, the fuel throughout the pipe line would have been at uniform pressure 1-2-3, and moving with uniform velocity towards the nozzle. Owing, however, to the increasing velocity of the pump plunger, the pressure at the first measuring point will be increasing along 3-4, the stepped nature of the pressure rise being due to vibrations set up in the diaphragm of the first measuring point. The velocity of the pump plunger will determine the general steepness of the wave front 1-2-3-4 which increases with speed as a comparison between the rate of pressure rise in Fig. 48 taken at 330 r.p.m. and that obtaining in Fig.49 recorded under similar throttle and injection-pressure condition but at a pump speed of 900 r.p.m. will serve to show.

On arrival at the nozzle, the initial velocity of the fuel corresponding to the pressure represented by 1-2-3 will suffer change according to the opening conditions of the needle valve, and the resulting change of momentum will build up corresponding changes in pressure which will be

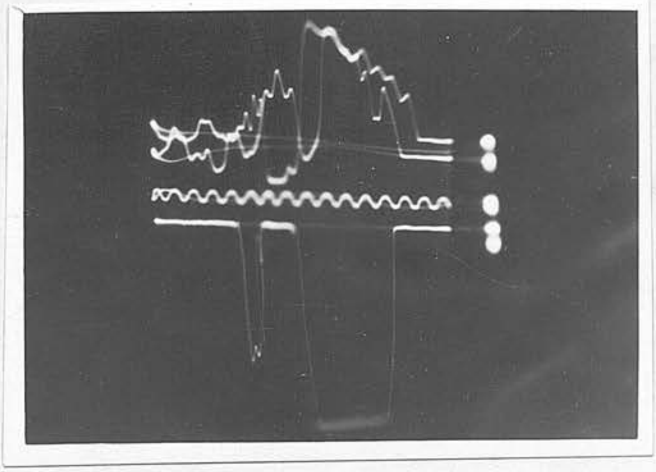


FIG. 49.

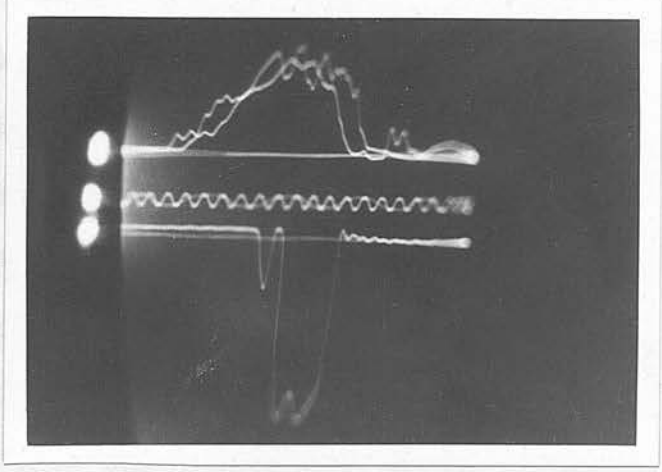


FIG. 50.

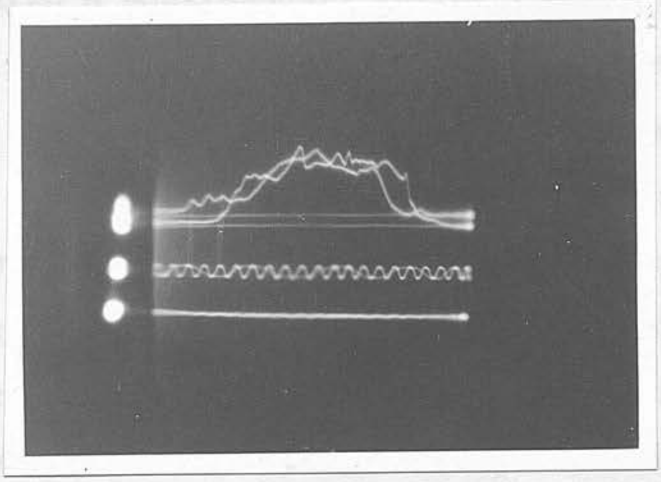


FIG. 51.

propagated towards the pump with the velocity of sound. If the needle fails to open under the initial impulse (as in the case under discussion) complete reflection of the wave front will result in a wave of 'zero' velocity with double the initial pressure intensity 1-2-3, being propagated towards the pump. This is clearly shown by a-b-c-d in the nozzle pressure record. Under other circumstances where opening of the needle takes place, reflection may be partial or may be negative in character resulting in the propagation of a wave of decreased pressure towards the pump.

With a continuously increasing plunger velocity (as with the present system) a true state of zero velocity cannot exist, since successive increments of the advancing sloping wave front 3-4 will suffer velocity change at the reflected wave front travelling towards the pump and the corresponding pressure changes transmitted forward to the pump and back to the nozzle, at which latter point, they will again experience velocity change and again build up changes of pressure (d-e in the nozzle diagram) which in turn are propagated towards the pump.

At point 4, the front of the reflected wave of 'zero' velocity emanating from 'a' at the nozzle has reached the pump plunger as indicated by the rapid pressure increase along 4-5 in the pump-pressure diagram. Receiving momentum from the plunger which is still moving with increased velocity, a reinforcing pressure wave is now propagated towards the nozzle, arriving there at point 'e' where as before its consequent pressure change resulting from change in velocity as determined by nozzle opening and operating conditions is propagated as a wave towards the pump arriving there at point 6 in the pump pressure diagram.

It will thus be seen that in an Injection System similar to that under consideration, the pressure at any point in the system after the passage of the initial disturbance is the result of plunger velocity and nozzle opening and operating conditions.

For a given pump speed and pipe length, the behaviour of the latter will be affected by the set opening pressure of the needle valve and the residual pressure in the fuel line as determining the intensity of the pressure necessary to overcome the needle tension and by the area of the orifice in relation to pump area as determining the rate of flow.

The opening of the needle valve at f corresponding to the point F in the valve lift diagram occurs under one of three conditions. In the first case, the intensity of the primary pressure impulse 1-2-3 may be of sufficient magnitude to open the needle, with little or no reflection and the total injection lag, between opening of delivery valve and the needle lift would closely correspond with the pressure propagation lag 1-a. Under the second condition of opening, the initial impulse is insufficient to overcome the needle tension and some reflection of the primary impulse (1-2-3) and possibly of the sloping wave front (3-4) becomes necessary. Under these conditions of opening, the injection lag would be the sum of the pressure lag and the time necessary to build up the higher pressure by reflection.

Lastly, as in Fig. 48 conditions may require the complete reflection of the wave front (1-2-3-4) (a-b-c-d-e) with subsequent reinforcement from the pump plunger (e-f) and in certain cases, the process of complete reflection and reinforcement may require repetition before a sufficiently intense pressure has been built up under the needle valve seat. Evidently the injection lag under this third method of needle opening will be the sum of the pressure lag 1-a, the time required for the necessary number of double passages of the pressure wave through the fuel line and possibly further partial reflection at the nozzle. In each of the cases considered, some small additional time will be required to overcome the inertia of the valve and its associated equipment.

From a study of the foregoing, it would appear that conditions remaining constant, the injection lag should decrease in time with increase of pump speed as both the primary pressure impulse and the subsequent reinforcement of the reflected waves increase with increasing plunger velocity. This feature is amply borne out by the present investigations. When the needle valve lifts at f in response to a pressure accumulation under the valve seat, a drop in pressure takes place as shown by f-g, this being due as mentioned previously to the increased capacity of the system by the enlargement of the pressure space in the jet plate and by the needle movement. The magnitude of the pressure drop evidently depends upon the rate at which the fuel is being extruded through the jet as affecting the change of velocity and thus the tendency to pressure drop, and the rate at which the pressure is at that instant, being built up at the valve due to the incidence, reflection or reinforcement of the pressure waves. This latter condition depends on the past history of the system as due to the limited velocity with which pressure or velocity changes can be propagated through the fuel, changing conditions at the pump cannot be manifested at the nozzle at the instant of opening of the needle valve.

The intensity of the initial impulse imparted to the valve causes it to lift as shown by FG in the valve lift diagram, the rapidity and extent of this initial lift being dependent upon the magnitude of the impulse. With the drop in pressure f.g. accompanying the lift FG, the needle valve tends to approach its seat with consequent throttling of the flow, and corresponding arrest in the pressure drop. With an opportunely advancing pressure wave from the pump or a sufficiently strong reflection of the previous wave, the needle may be held in the open position and flow take place with diminished velocity. This is the case in Fig. 48 where the reflection of the wave 4.5 from the pump is being effected when the needle lifts at f and flow takes place under the pressure gh. as shown by GH in the valve lift record. Under other circumstances, the intensity of the initial impulse may be sufficient to raise the valve to its full lift in which position it is maintained by its inertia whilst oncoming or reflected pressure fronts are enabled to build up pressures greater than the closing pressure and so hold the valve at its full lift throughout the injection. Such a case is well illustrated in Fig. 49. On the other hand, the drop in pressure fg accompanying the lift may be so great and the rate of building up at the instant of lifting so small, that the valve reseats and remains closed, until forced to lift again by the reinforcing waves from the pump. These conditions are shown in Fig. 50 which refers to the 2 feet pipe with a pump speed of 330 r.p.m. and a nozzle setting of 150 atmospheres. The pressure necdssary to cause reopening is not necessarily that for which the needle valve is set as the resulting bounce when the valve strikes its seat on closing, exposes the whole valve area to the acting pressure in place of the differential shoulder which is exposed when the valve is in its closed position. For the same reason, the closing pressure of the needle valve is always less than that of opening.

It will now be advisable to consider the conditions at the pump in relation to the changing conditions at the nozzle. At the instant of theneedle opening at f ,the front of the previously reflected wave from the nozzle (ef) has arrived at the pump at point 6 and pressure is being built up as previously explained along the front 6,7. That part of the pressure rise along 6-7 due to the decrease in velocity of the front 4-5-6 as it meets the advancing reflected front e.f. emanating from the nozzle at 1 has so far depended on a stationary condition of the fuel at the nozzle. Initiation of velocity in the needle jet at f calls for a corresponding velocity in the pipe line and a corresponding drop of pressure which, as before, is propagated along the pipe towards the pump with approximately the velocity of sound in the fuel. Its arrival at the pump is manifested by the drop 7-8.

Arising from previous discussion, it will be seen that had the nozzle remained closed, the pressure condition at the nozzle would have assumed the form shown approximately by ff'f" and the corresponding conditions at the pump by 77'7". The effect of the negative pressure wave emanating from the nozzle following the lift of the needle will now be apparent. The strong wave front eff' which would have arrived at the nozzle has been reduced to the weak pressure characteristic shown by efgh, and the positive reinforcing wave front 677'7" at the pump replaced by a negative front 6789.

The weak reflection gh at the nozzle is responsible for the lift represented by GH on the valve lift diagram and the arrival of the reinforced wave front 677'7" which under closed condition, would have appeared as at f', now greatly reduced, appears at h and the lift is increased more rapidly as shown by HH' on the lift diagram. But the resultant increased capacity and velocity is more than the reinforcing wave from the pump can support and the valve moves towards its seat under the influence of its spring as shown by H'K and continues to do so until it again receives the weak reinforcement from the pump at k as shown at point K in the lift record.

Returning to conditions at the pump, it will be seen that the effect of the increased lift of the needle as shown by G.HH' is to produce a velocity in the pipe line of such an order that the consequent pressure reduction carried towards the pump is greater than the increased pressure which would normally accrue from the ever increasing velocity of the pump plunger, and the algebraic sum of these two influences results in a net decrease as shown by the sloping front 789.

With the decreasing lift H'K however, weak reflections are built up at the nozzle as the slightly increasing pressures noted at hk will serve to verify. These increases are transmitted towards the pump and, receiving reinforcements from the plunger whose velocity is now greater than the velocity of the fuel at the nozzle builds up a slightly increasing pressure front as shown by 9.10.

Shortly before the point 10, the plump plunger has registered with the suction port in the pump barrel and the pressure reduced to atmospheric at that point. The first effect of the release is to bring the fuel in the pipe to a state of rest at a pressure corresponding to that existing before release. A wave of zero velocity is therefore propagated along the pipe towards the nozzle to be immediately followed by a pressure-depression and negative velocity wave as the fuel expands through the suction port and the pipe unloaded by the unloading valve (10-11-12).

Several factors combine to determine the magnitude of this wave. The speed with which the suction port is increased is the first factor of importance as it determines the rate at which the fuel in the pipe can expand to atmospheric pressure, and hence influence the slope of the depression wave.

Again the intensity will be determined by the type of unloading valve peculiar to the system. In that under consideration whereby not only is the fuel at the pump broughtto rest but actually given a negative velocity due to the unloading section incorporated in the valve, much depends on the rate of closing of the valve and resultant magnitude of the negative velocity. With the rapid uncovering of the suction ports at high speeds and large throttle openings, it is to be expected that the closing will be rapid, and the slope of the negative wave become steeper. At the lower pump velocities, where the speed of delivery valve closing is greatly reduced, the unloading effect is correspondingly decreased.

To compare the release effects at low speeds between the unrelieved and unloaded systems, the delivery valve was removed and records taken of the pressure characteristics. It was found that at all throttle openings and pipe lengths up to a speed of 385 r.p.m., the effect of the unloading valve as influencing the rate of cut-off was practically negligible, as a comparison between Fig.51

taken under the conditions exactly similar to Fig. 48, but with the delivery valve removed will serve to illustrate.

The first effect of the depression wave reaches the nozzle at point $\ell$ and follows the line $\ell m$. It will be noted that owing to the closing of the needle and resulting decreased capacity, the slope of the depression $\ell m$ is less steep than that at the pump and is actually the algebraic sum of the two opposing effects. With the decreasing pressures, the needle valve spring asserts itself and the valve is forced towards its seat as shown by the closing line LM. At M, the needle valve closes and a momentary building up of pressure results as indicated by the rise mn in the nozzle pressure diagram. With continued intensification of the negative wave 10-11, the pressure at the nozzle continues to fall as indicated by nop.

In the meantime, the pressure at the pump continues to decrease as indicated by 10-11-12. At the point 12, the delivery valve closes and with the sudden stoppage, a building up of pressure takes place as indicated by the rise 12-13. The fuel line between delivery valve and needle is now a closed system, and since no energy is being received from outside sources, a pressure increase at the pump will be accompanied by a pressure deficit at the nozzle. There are thus set up in the pipe-line fuel-oscillations of a more or less intensive nature according to the pressure conditions at release, the duration of opening and rate of closing of the delivery valve. The oscillations take place with a frequency given by $f = \frac{4L}{V}$ where L = Pipe-Length in feet, and V the velocity of sound in the fuel. Berg & Rode[12] have noted that these oscillations occur with frequencies corresponding to velocities other than that noted above and varying with the length of the pipe. The present investigations do not confirm their findings.

Under certain circumstances of pump speed, throttle and nozzle setting, the oscillations may be so severe as to cause reopening of the needle valve. Such reopenings are facilitated by needle-bounce exposing the whole area of the needle to the pressure accumulations, and may thus take place at pressures considerably less than those required to overcome the set needle tension. Post injections are detrimental to efficient engine performance, as, entering the cylinder, when the available oxygen has already been largely utilised, they cause late burning with consequent overheating and smoky exhaust. To illustrate the phenomena of post-injection, the tension of the delivery-valve spring was considerably increased above the normal setting on the assumption that the increased tension would cause the delivery valve to close quickly following pump pressure release, thus ensuring the initiation of high fuel velocities.

With this alteration, post-injection phenomena were observed to take place at all pump speeds between 900 r.p.m. and 450 r.p.m. with the three pipe-lengths examined. The number of post-injections occurring under the conditions examined are outlined in Table II, and Figs. 52. to 54 taken under the conditions noted over a pump sweep of 120° show typical records of the injection phenomena. Under normal working conditions of the system investigated, post-injections do not occur and the phenomena have therefore not been extensively studied in the present work.

It is evident however from a study of Table II that post-injection takes place more freely at higher speeds and throttle openings and at lower needle valve settings, and that the pipe-length is a contributing factor as evidenced by the greater number of double post-injections taking place with the four-feet pipe.

TABLE II

| LENGTH. | SPEED | THROTTLE | NUMBER OF POST-INJECTIONS. | | |
|---|---|---|---|---|---|
| | | | 100 KG per $CM^2$ | 150 KG per $CM^2$ | 200 KG per $CM^2$. |
| 8' | 900 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>2 | 1<br>1 | 1<br>1 |
| | 750 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>1 | 1<br>1 | 1<br>1 |
| | 450 | $\frac{3}{4}$ | 1 | 1 | 1 |
| 4' | 900 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>2 | 1<br>2 | 1<br>1 |
| | 700 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>2 | 1<br>1 | 1<br>1 |
| | 450 | $\frac{3}{4}$ | 1 | 1 | 1 |
| 2' | 900 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>2 | 1<br>1 | 1<br>1 |
| | 700 | $\frac{1}{2}$<br>$\frac{3}{4}$ | 1<br>1 | 1<br>1 | -<br>1 |
| | 450 | $\frac{3}{4}$ | 1 | - | - |

A broad survey of the injection process reveals that the injection characteristics obey a law of supply and depletion; the pump at any instant delivering a certain amount of fuel which the nozzle some time later is called upon to deliver. The conditions affecting delivery at this

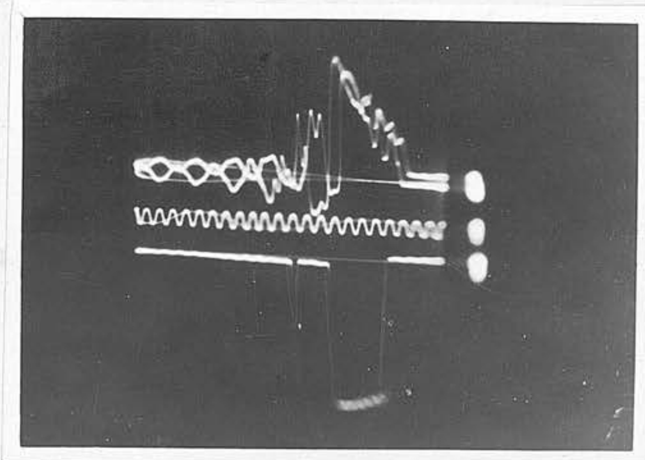


FIG. 52.

2Ft.750 R.P.M. $\frac{1}{2}$ THROTTLE.

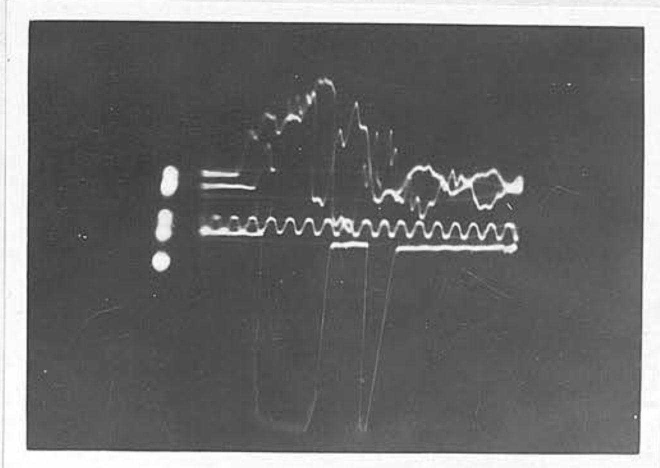


FIG. 53.

4Ft.950 R.P.M. $\frac{1}{2}$ THROTTLE.

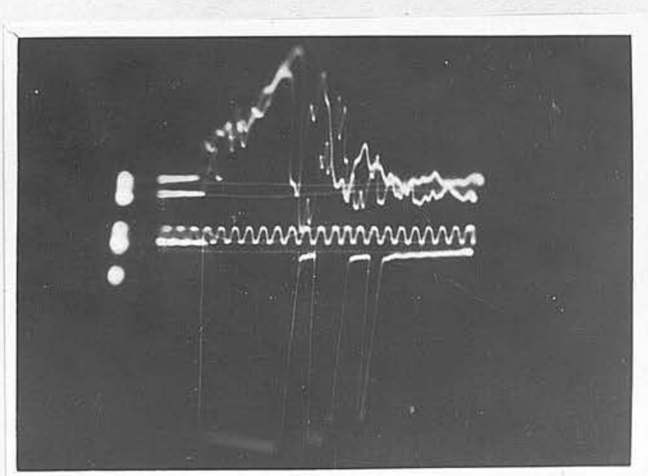


FIG. 54.

4Ft.950 R.P.M. $\frac{3}{4}$ THROTTLE

100 KG.per CM$^2$

instant are largely based on the past history of the pump's movements and the nozzle may be in a position to extrude a greater quantity than it is receiving or alternatively, the supply may be greater than the nozzle can pass. The former conditions result in reduced pressure throughout the system, the latter in rising pressures, the tendency at all times being to build up those conditions which will allow of the depletion from the nozzle to equal the supply from the pump.

PART IV.

EXPERIMENTAL RESULTS.

SECTION III.

ANALYSIS of RECORDS.

(i) Detailed Analysis.
(ii) General Analysis.

## ANALYSIS of PRESSURE and NEEDLE-LIFT RECORDS.

In a subsequent section of the work, an analysis is made of the injection characteristics as revealed by measurements taken from the records. The present section gives a brief survey of certain features of the injection process readily obtained by visual examination, and to facilitate comparison of the records under the varying speed and load conditions employed in the investigations, the diagrams have been arranged in a separate folio accompanying this thesis. These are arranged under major headings of pipe length with subdivisions according to needle setting. Each sheet of records is further sub-divided according to speed and throttle opening conditions.

To serve the same purpose, the needle lift records have been sketched in Fig. 55, which shows the salient features discussed in the following section of the work.

### (i) Detailed Analysis of Records.

(1) 200 r.pm. (Nominal).

(a) 2 Feet Pipe : 100 KG/CM$^2$. The initial pressure impulse has been twice reflected from the nozzle with subsequent reinforcement at the pump before the needle lifts. The lift, coinciding with the arrival of the reinforcing wave, a strong opening, results. Following the initial discharge, a negative wave is propagated to the pump and at the low pump speed although the velocity of the pump plunger is increasing, the pressure deficit cannot be made up. Accordingly, the pressure falls as indicated by the drop in the pump-pressure curve.

When the rate of discharge decreases due to pressure deficiency at the nozzle (shown by the drop of the needle towards its seat) weak reflections from the nozzle are reinforced at the pump and the pressure rises, causing an increased rate of flow indicated by increased needle lift. Following the increased velocity through the nozzle a negative wave is again propagated towards the pump and the process outlined above repeated.

Pressures in the system therefore follow a course dictated by needle movements and injection takes place under decreasing pressures as shown by the general slope of the pressure and needle-lift diagrams, the maximum pressure in the system being that obtaining at initial needle lift.

<u>150 and 200KG/CM$^2$</u>: With needle tension set at 150KG per CM$^2$ opening also takes place following the arrival of the second reinforcing wave from the pump but with a further increase to 200KG per CM$^2$ a third reinforcement is necessary. The opening is weaker than was the case with the lower needle setting and with increasing needle tension, the discharge characteristics

while following the same trend as those occurring with the lower needle setting divide into more clearly defined periods of injection and closure. With a gradually decreasing pressure characteristic, the duration of successive openings becomes less and the time of closure increases as injection proceeds.

A feature of the low speed high needle setting records is the relatively large residual pressure in the fuel line between injections. This may be explained in terms of the needle movement, which, on account of the weak pressure reinforcement never lifts to its full travel but flutters near its seat. On arrival of the front of the depression wave following release the needle will instantly seat thus sealing the fuel pipe at the nozzle-end at a relatively high pressure.

(b) <u>4 Feet Pipe</u>: <u>100 KG/CM$^2$</u> . The pressure characteristics are similar to those for the shorter pipe but it will be noted that due to the increasing residual pressures which occur with increasing throttle, the needle valve at the $\frac{3}{4}$ setting opens with the partial reflection of the sloping wave front following the first reinforcement from the pump, as compared with the double reinforcement necessary with the $\frac{1}{4}$ and $\frac{1}{2}$ nozzle settings. Lacking reinforcement, the lift is of short duration and the valve closes until the advancing reinforcing pressure wave from the pump causes re-lift. Thereafter, the pressure and discharge characteristics follow closely those obtaining with the two feet pipe.

150 and 200 $KG/CM^2$. With the needle set to open at 150 KG per $cm^2$, opening is initiated with the partial reflection of the sloping front following the first reinforcement while at the 200 $KG/CM^2$, setting, injection begins with the arrival of the second reinforcement. A breaking up of the injection similar to that occurring in the two feet pipe at the same speed takes place at the higher needle settings.

(c) 8 feet pipe: 100KG per $CM^2$: Owing to the longer time available for the reflection of the sloping front of the initial wave from the pump, the needle valve opens soon after the arrival of the first reinforcement from the pump and with higher residual pressures at the ¼ and ¾ throttle settings, the necessary reflection of the initial reinforcement is less than is the case with the ½ throttle setting, where almost complete reflection is necessary to initiate opening. At the lower speeds investigated, the residual pressure does not necessarily increase with increased throttle owing to the powerful influence of the needle movements on the fuel line pressures. In the present instance, the arrival of the depression wave at the nozzle following upon release, coincides with a state of increasing pressures with the ¼ and ¾ nozzle settings and with a decreasing pressure at the ½ nozzle setting.

An indication of the residual pressures is always given by the nature of the pendulations in the line following the closing of the delivery valve, since their intensity is dependent upon the velocities set up with release of strain energy in the fuel and is thus indicative of the mean strain energy existing in the fuel, when, after damping of the waves, it assumes a state of rest.

150 AT. and 200 AT. At the 150 At. and 200 At. needle settings, injection begins with the reflection of the first reinforcing pressure wave from the pump, the necessary reflection being less with both settings at the ½ and ¾ throttle openings than with the four-feet pipe due to the higher residual pressures obtaining in the fuel line. Injection at the highest needle tension with the eight feet pipe is spasmodic varying from cycle to cycle.

(2) 350 r.p.m. (Nominal)

(a) 2 Feet Pipe. 100 KG/CM². The needle opens with the almost completed reflection of the first reinforcing wave from the pump. There is consequently a large pressure drop following valve lift and in the case of the half throttle setting, the needle almost reseats before the rise in pressure consequent upon closing together with the weak reinforcing wave from the pump is sufficient to cause increased lift. Injection takes place under decreasing pressure until near the end of the ½ throttle delivery when pressure reflection in conjunction with ever increasing pump plunger velocity combine to give an increased rate of injection under rising pressures.

150 and 200 KG/CM$^2$: The needle lifts after two reinforcements when set to 150 atmospheres and after the complete or partial reflection of the sloping wave front following the primary wave, at the 200 atmosphere setting, the reflection being complete at the ¼ nozzle setting and partial at the ½ and ¾ settings. In common with the four feet pipe, injection takes place under a slightly falling pressure characteristic.

(b) 4 Feet Pipe: 100, 150 and 200 KG per CM$^2$. The pressure characteristics follow closely those for the two feet pipe but owing to higher residual pressures in the fuel line, the injection is better sustained and the pressure characteristics less sensitive to needle movements.

(c) 8 feet Pipe: 100 KG per CM$^2$. Opening of the needle takes place with the partial reflection of the sloping front of the primary wave, and until the arrival of the reinforcement, violent fluctuations in needle lift follow the initial opening. Needle movements are strongly reflected in the fuel line pressures existing at the nozzle but owing to the circumstances attending the opening of the needle are not reflected to the same extent at the pump.

The weak opening due to the reflection of the sloping wave front causes weak negative reflections to be propagated towards the pump but with the long pipe, the ever increasing velocity of the pump plunger has sufficient time to assert itself, the effect of the negative reflection is overcome and a positive reinforcement results. Following the poor opening, a strong well-sustained discharge ensues under a rising pressure characteristic. It is noteworthy that the conditions of injection in the 8 feet pipe for the three needle settings investigated at a pump speed of about 350 r.p.m. are the only ones in the whole series of variable conditions investigated, which enable the maximum pump pressure to rise appreciably above that at the nozzle.

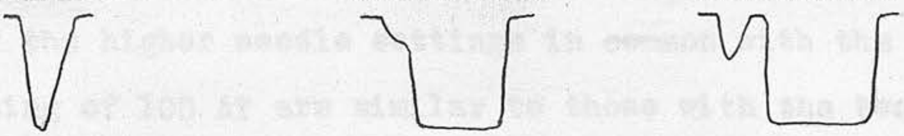
<u>150 and 200 KG per CM$^2$</u>. With the exception of the $\frac{1}{4}$ throttle setting with 200 atmospheres needle tension where injection follows the reflection of the primary wave, needle-opening occurs with the arrival of the first reinforcement from the pump. As the throttle is increased to the $\frac{3}{4}$ setting, the pump speed has risen sufficiently to build up reinforcements from weak positive reflection from the nozzle to a degree enabling a rising pressure characteristic

to be maintained. A well sustained, end-injection results under these conditions, cut-off being sharp and clean.

(3) 450 r.p.m.

(2) 2 Feet Pipe: 100 KG/CM$^2$. The partial reflection of the initial sloping wave front from the pump is sufficient to induce needle lift, the necessary reflection decreasing with increased throttle setting due to increasing residual pressures. The corresponding increase in the power of the initial wave to initiate needle opening is clearly shown in the valve lift records roughly reproduced below.

½ Throttle. ⅝ Throttle. ¾ Throttle.



The initial lift still results in a negative wave now of greatly reduced intensity being propagated towards the pump but the increasing velocity of the plunger thereafter maintains an increasing pressure and, as may be observed from the record for the ¾ throttle setting, reflections and reinforcements occur in a manner similar to that taking place before needle-lift but with a reduced intensity as indicated by the more gradual slope of the pressure records.

150 and 200 KG/CM$^2$. With the needle setting

increased to 150 AT. injection begins with partial reflection of the first reinforcement and similar opening conditions obtain with increase of needle tension to 200 atmospheres. In common with the lower setting, a rising pressure characteristic results from the increased plunger velocity and excellent discharge characteristics obtain throughout the whole range of load conditions examined.

(b) 4 Feet Pipe. 100 KG. per $CM^2$. The opening conditions are similar to those for the two feet pipe, but with slightly increased residual pressures, the characteristics of the latter pipe are accentuated.

150 and 200 KG. per $CM^2$: The opening characteristics for the higher needle settings in common with the lower setting of 100 AT are similar to those with the two feet pipe, and both pressure and needle lift follow closely the corresponding characteristics for the shorter pipe.

(c) 8 Feet Pipe. 100 KG. per $CM^2$. Opening with the partial reflection of the sloping wave front following the initial impulse, fluctuations, in discharge rate ensue until the arrival of the reinforcement causes an increased needle lift and rate of injection. A weak negative reflection following initial opening is followed by weak positive reflections and stronger pump reinforcements result in an increasing pressure front.

<u>150 and 200 KG per CM$^2$</u>: With needle setting increased to 150 KG per CM$^2$, injection begins at the $\frac{1}{4}$ throttle setting with the reflection of the initial sloping wave front. Closure of the needle quickly follows initial opening, but the slowly increasing pressure due to further wave front reflection causes a weak re-opening before release occurs. The $\frac{1}{2}$ and $\frac{3}{4}$ throttle records show the same opening characteristics intensified as a result of higher residual pressures. Thereafter the arrival of the reinforcement from the pump causes a strong re-opening of the needle and injections terminate with increasing pressure characteristics. Similar characteristics are observable with the needle tension set to 200 atmospheres.

(4) <u>700 and 900 r.p.m.</u>

<u>100, 150 and 200 KG per CM$^2$</u>: With the increase in pump speeds above 500 r.p.m., injection takes place following partial reflection of the primary wave. Following needle opening, strong reflections and reinforcements serve to build up a rapidly rising pressure characteristic showing close correlation with pump plunger velocity.

In no case however, does the pressure drop following valve-lift fail to be transferred as a negative reflection to the pump. The finding of Berg and Rode[12] that as a general rule, the pressure drop propelled towards the pump is completely flattened out by the fluid and pipe friction is not confirmed in the present investigations.

With both increasing speed and throttle opening, the depression wave becomes steeper and at 900 r.p.m. with the nozzle at the $\frac{3}{4}$ setting becomes almost vertical. Fuel oscillations in the tube occur with intensities increasing with increased speed and larger throttle but in no case are they of sufficient magnitude to cause reinjection.

A feature of the pressure records particularly noticeable at the higher speeds is a peculiar sudden drop in pressure in the pump pressure record during the early part of the delivery stroke. The drop does not occur at a fixed time interval following the start of pressure rise registered at the measuring point near the pump, but takes place at time intervals varying with pump speed. The occurrence may therefore be dissociated from peculiarities of the measuring system.

As will be evident from the plunger lift and velocity curves in Fig. 38, no discontinuity occurs. Careful examination of the pump plunger and barrel showed no defects likely to cause leakage at this period.

The writer is therefore of the opinion that the phenomenon is associated with the lift of the delivery valve as being the only other variable in the system capable of showing such a drop varying with pump speed. When the arrival of this pressure drop at the nozzle coincides with needle lift, partial closing of the needle takes place followed of course by a strong opening as the pressure defect is overcome by the increasing reinforcement. This feature is particularly noticeable at the 750 speed range for the higher needle settings.

(ii) General Analysis.

At the lower speeds owing to weak primary waves, strong negative reflections from the nozzle and weak reinforcements from the pump, needle opening pressures are not maintained, and the rate of discharge fluctutes. The speed at which such fluctuations become marked, varies with pipe length and with needle valve setting. At the speed of 205 r.p.m. and needle setting of 150 and 200 atmospheres, the needle reseats following initial opening, due to pressure deficiency followed by repeated re-opening and closing in response to pressure accumulations, and depletions. At the same speed, but with the needle set to 100 atmospheres, while fluctuations in discharge rate are evident in each of the three pipe lengths investigated intermediate reseating and reopening of the needle before final closure does not occur.

The degree of fluctuation increases with both increase of needle tension and pipe length and in the case of the 200 atmosphere setting at a speed of 205 r.p.m., the discharge characteristics with the eight feet pipe were not consistent but varied from stroke to stroke of the pump.

With a pump speed of 350 r.p.m. discharge fluctuation persist and are characterised by a weak opening, a reasonably well sustained intermediate period and weak termination. The fluctuations increase with increasing needle tension and pipe length. When the speed of the pump is increased to 450 r.p.m. the character of the discharge changes to that of a rather weak opening followed by a well sustained injection, which persists throughout the discharge period, the injection terminating in a sharp clean manner. The weak opening is intensified with increasing needle tension, and owing to the relatively long period between reinforcements, the influence of needle movements, now small with the shorter pipes, still shows to a large degree in the eight feet pipe. The higher pump speeds of 700 and 900 r.p.m. give generally desirable injection characteristics consisting of a sharp opening, a sustained period under increasing pressure ensuring good penetration and atomisation and a clean termination. This applies to all needle tensions and pipe lengths with a few exceptions notably at the pump speed of 700 r.p.m. With increasing speeds, the possibility of adverse injection rates obtaining is minimised due to the stronger impulses from the ump, but certain conditions may exist depending on residual pressure, pump speed, pipe length, throttle and needle-tension, whereby the initial opening of the needle is accompanied by a pressure drop of such magnitude that the oncoming wave is unable to immediately make up the deficiency. Neglecting these minor variations, it may be stated that within the scope of the injection conditions investigated discharge fluctuations likely to be detrimental to efficient engine operation cease at a pump speed of about 450 r.p.m.

Unfavourable opening conditions of discharge persist throughout a greater range of speed than defective closing characteristics. With isolated exceptions, terminal injection under low or fluctuating pressure does not occur at pump speeds greater than 350 r.p.m.

At the lowest speed investigated, the injection takes place under a falling pressure characteristic, the maximum pressure throughout the injection being that at needle lift. When the pump speed is increased to 450 r.p.m. the pressure reinforcements from the pump maintain a slightly rising characteristic and at the higher speed ranges pressure at pump and nozzle rise rapidly with increase in speed and show close correlation with corresponding pump plunger velocities. In general, the maximum pump pressure is less than the maximum at the nozzle with the notable exception of the

8 feet pipe at 350 r.p.m. In this particular case, as previously noted, the maximum pump pressure is greater than that at the nozzle for the three needle settings investigated.

The discharge and pressure characteristics for a given pipe length and pump speed at the maximum throttle opening are largely maintained at lower throttle openings under the same speed and load conditions. This is particularly noticeable at the lower speeds and low needle settings where residual pressures are comparatively small and cause little variation in the absolute values of the pressures in the fuel line. Under these conditions, the effect of decreasing throttle is as it were to cut off portion of the full load characteristic. In the records for the 2 feet pipe for a pump speed of 350 r.p.m. reproduced in Figs. 56 and 58, this feature is particularly well brought out.

It may be further noted that the discharge characteristics for a given pump speed and pipe length tend to persist with both increase in needle setting and pipe length. Under such conditions, discharge fluctuations occurring with the lower needle setting and shorter pipe are intensified as is readily observable from a study of the needle lift diagrams in Fig. 55. (Folio).

At the higher pump speeds, partial or complete reflection of the primary wave is sufficient to cause needle lift for all conditions of needle setting and throttle variations investigated. As the speed is decreased, increasing

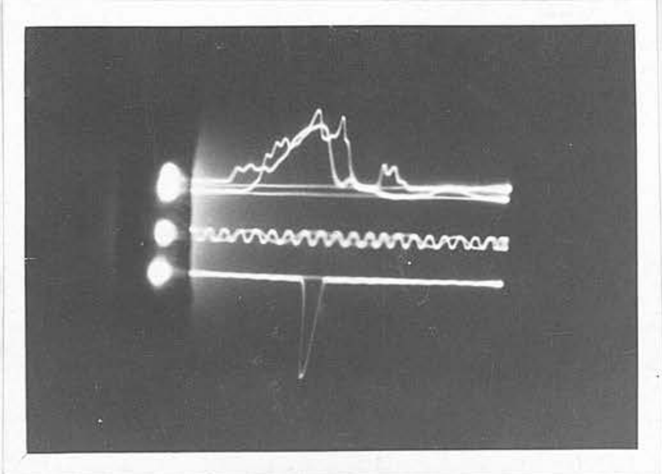


FIG. 56.

2Ft. 330 R.P.M. 100 KG PER CM.$^{2}$

$\frac{1}{4}$ THROTTLE.

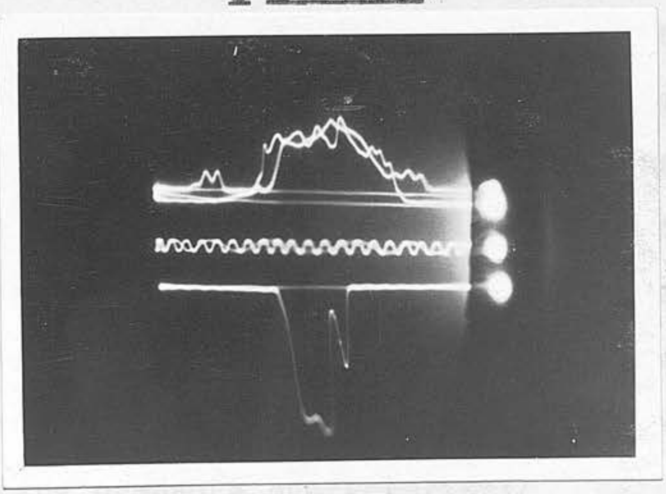


FIG. 57.

$\frac{1}{2}$ THROTTLE.

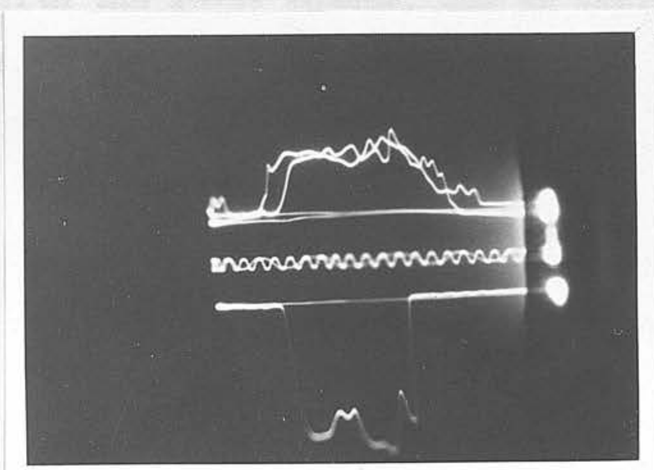


FIG. 58.

$\frac{3}{4}$ THROTTLE.

reinforcement of the primary wave becomes necessary to initiate discharge, the number of reinforcements increasing with decreasing pipe length.

At all speeds investigated, the pressure drop consequent upon initial needle lift is transferred as a negative reflection to the pump and in all cases, influences the slope of the pump pressure characteristics. At the pump speeds above 450 r.p.m. increasing pump plunger velocity enables the reinforcements to overcome the pressure depletion and a rising pressure characteristic results; the slope increasing with increasing speed. As the speed falls below 500 r.p.m. however, the effect becomes increasingly felt and at a pump speed of 200 r.p.m. injection takes place under a falling pressure characteristic. At the higher speeds, the pressure reflection and reinforcement necessary to cause needle lift decreases with speed and throttle increase. With the lower speeds, due to variation in residual pressure, the same consistency is not observed.

# EXPERIMENTAL RESULTS.

## SECTION IV.

(i) Measurement of Records.

(ii) Explanation of Tables.

(iii) Discharge Characteristics.

(i) SECTION IV. EXPERIMENTAL RESULTS.

While useful information regarding the injection process may be obtained from visual examination of the records, it was realised at the outset of the investigation that unless specific values could be assigned to the more important features of the characteristics, much of the possible usefulness of the work would be lost.

Accordingly, throughout the experiments every care has been exercised in maintaining constancy of recording conditions with a view to enabling a common scale to be applied to the results. With so many variable factors influencing both the operation and recording small variations over an extended series of experiments are inevitable but frequent checking throughout the work showed close agreement of recorded results.

The measurements recorded in Tables 3 to 11 were made by traversing the negatives on a specially adapted micrometer stage of a Zeiss Metallurgical Microscope reading to 0.1mm., observations being made through a No. 4 ocular with cross-hairs and a Zeiss Planar 1:4.5 R. 3.5cm. objective. These conditions of observation gave a reasonable magnification without unduly destroying the distinctive outlines of the diagrams.

The Tables 3-11 are arranged under the respective pipe-lengths and needle settings and the data is set out against speed, in groups corresponding to nominal speeds of 900, 700, 450, 350 and 200 revolutions per minute respectively.

A short explanation of the Tables is given in the following Section.

## (ii) EXPLANATION of TABLES.

Speed: With cyclic and voltage variations occurring in the mains, motor speed varied above and below the nominal setting. The actual pump speed at the instant of taking the record was obtained as previously described by taking the record over a fixed angular sweep of the pump, and from the time-base, determining the corresponding time in milli-seconds. For the two higher speed ranges, the sweep was set to $80^{o}$ and for the remaining ranges to $60^{o}$ of pump rotation.

Throttle: The settings designated $\frac{1}{4}$, $\frac{1}{2}$ and $\frac{3}{4}$ throttle do not refer to the fraction of the full charge, but to the position of the throttle control rod relatively to its full travel. The full charge on the pump investigated corresponds to a travel of the control rod of 24 mm. from the "Stop" position.

Scale of Time: Changing pump speeds required alteration of the velocity control to give a reasonable length of diagram. It was not possible therefore to work to a fixed setting and the time-scale in millimetres per millisecond varies with changing pump speed.

Lag: The lag is a measure of the time taken for the initial pressure impulse to traverse the pipe between the pressure-measuring points. It does not indicate the time lag between closing of pump suction port and the arrival of the pressure impulse at the nozzle as the plunger will have moved through portion of its delivery stroke compressing the fuel in the pump barrel to that existing in the pipe beyond the delivery valve before the first pressure rise is noted at the first measuring point.

The distance measured is greater than the nominal lengths of the pipes tested by the sum of the distances between the pipe ends and the diaphragms of the measuring points. The actual distances over which the lag is measured are:-

| | | |
|---|---|---|
| With 2 Feet Pipe | : | 33.175 inches. |
| With 4 Feet Pipe | : | 57.175 " . |
| With 8 Feet Pipe | : | 105.175 " . |

All time measurements are made taking the point of initial pressure rise at the first measuring point as zero time.

<u>Velocity of Propagation of Pressure Wave</u>: These columns are determined from the lengths noted above and the measured time lag.

<u>Injection Lag</u> (<u>Milli-seconds</u>). Figures in this column express the injection time-lag measured between the first indication of pressure rise at the indicator near the pump and needle lift. The pressure rise at the first measuring point in relation to closure of the suction port by the plunger measured in degrees of pump angle showed such slight variation throughout the speed and load range, and as absolute values were not sought, special arrangements to measure the differences were not adopted.

The values of injection lag (and also cut-off lag) are therefore only comparative within an accuracy dictated by variation in residual pressures.

The capacity of the pump chamber between suction port and delivery valve is a little in excess of 450mm.[3] and taking a compressibility coefficient of 1/10.000 per atmosphere, the maximum variation throughout the speed and load range investigated does not exceed one degree of pump revolution.

<u>Injection Lag</u> (<u>Degrees</u>). The Injection Lag in degrees is obtained from the measured injection lag in milli-seconds and the corresponding pump speed in the first

column.

<u>Valve Opening in Milliseconds and Degrees</u>. The opening is measured in millimetres on the valve lift diagram and from time-scale and speed data converted to milliseconds and degrees.

<u>Cut-Off Lag</u> (<u>Milli-seconds and Degrees</u>). These columns give the time interval between pump release and needle closure. To determine the release point in relation to the initial rise of pressure indicated on the pump-pressure records, the throttle was set and the phase selector of the mechanical time-sweep adjusted until the first pressure rise coincided with initiation of the spot's excursion across the screen. On slowly rotating the pump by hand, and noting its angular position at the instant the spot begins to move, the above condition on the screen becomes a known point in relation to pump rotation. With continued slow motion of the pump, the point of release was clearly felt and the angular position of the pump again noted. The difference between the readings gave the number of degrees between arrival of the pressure wave at the first measuring point and the point of release for the given throttle setting.

Maximum Nozzle Pressure and Residual Pressure: The maximum pressures occurring in the nozzle pressure records were measured in milli-metres and from the pressure calibration graphs (Fig. 47) the corresponding readings in KG per CM$^2$ obtained. The same procedure applies to the residual pressures. Owing to the unequal biassing of the tube plates, the time and pressure axes of the records are oblique and not rectangular. Pressure measurements were therefore made parallel to the axis obtained by drawing a line through the centres of the stationary spots as indicated in Fig. 48.

Tables 12-14: The discharge in cubic millimetres per pump stroke for the variable operating conditions examined are recorded under headings of pipe-lengths. The measurements were taken under the conditions outlined in Part IV Section I (ii) and represent the mean of four close determinations.

TABLE 3.

2 FEET PIPE - 100 KG PER $CM^2$

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT.PER SEC. | INJECT. LAG. MILLI-SECS. | INJECT. LAG. DEGREES. | NEEDLE OPEN. MILLI-SECS. | NEEDLE OPEN. DEGREES. | CUT-OFF LAG. MILLI-SECS. | CUT-OFF LAG. DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$. | RESIDUAL PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 905 | ¼ | 3.4 | .765 | 3607 | 1.47 | 8 | 2.292 | 12.45 | 2.11 | 11.45 | 175 | 19 |
| 913 | ½ | 3.37 | 0.618 | 4465 | 1.33 | 7.28 | 4.08 | 22.4 | 2.50 | 13.68 | 234 | 36 |
| 913 | ¾ | 3.43 | 0.578 | 4774 | 1.214 | 6.65 | 5.16 | 28.3 | 2.35 | 12.9 | 278 | 53 |
| 682 | ¼ | 2.67 | 0.975 | 2830 | 1.795 | 7.35 | 2.735 | 11.2 | 2.33 | 9.55 | 166 | - |
| 682 | ½ | 2.71 | 0.769 | 3588 | 1.69 | 6.91 | 5.04 | 20.6 | 2.81 | 11.51 | 202 | 22 |
| 705 | ¾ | 2.69 | 0.620 | 4450 | 1.63 | 6.90 | 6.20 | 26.2 | 2.62 | 11.1 | 266 | 44 |
| 417 | ¼ | 2.44 | 1.024 | 2695 | 4.78 | 11.98 | 2.01 | 5.04 | 3.34 | 8.35 | 164 | - |
| 417 | ½ | 2.61 | 0.72 | 3832 | 3.67 | 9.2 | 5.68 | 14.2 | 3.07 | 7.40 | 203 | 20 |
| 410 | ¾ | 2.44 | 0.60 | 4599 | 2.39 | 5.9 | 9.30 | 22.9 | 2.76 | 6.80 | 211 | 22 |
| 330 | ¼ | 2.08 | - | - | 5.92 | 11.5 | 1.45 | 2.86 | 2.70 | 5.36 | 158 | - |
| 330 | ½ | 2.06 | - | - | 5.00 | 10 | 5.55 | 11 | 2.53 | 5.00 | 164 | - |
| 330 | ¾ | 2.10 | - | - | 4.0 | 7.8 | 10. | 19.7 | 2.78 | 5.50 | 166 | 11 |
| 205 | ¼ | 1.3 | - | - | 8.42 | 10.34 | 2.81 | 3.46 | 3.9 | 4.80 | 145 | - |
| 205 | ½ | 1.3 | - | - | 8.02 | 9.86 | 7.45 | 9.16 | 2.45 | 3.02 | 150 | 11 |
| 205 | ¾ | 1.3 | - | - | 7.54 | 9.27 | 12.33 | 15.2 | 2.0 | 2.47 | 162 | 16 |

TABLE 4.

2 FEET PIPE - 150 KG. PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE. M.M. PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT. PER SEC. | INJECT. LAG. MILLI-SECS. | INJECT. LAG. DEGREES | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN DEGREES | CUT-OFF LAG. MILLI-SECS. | CUT-OFF LAG. DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$ | RESID. PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 890 | 1/4 | 3.94 | 0.53 | 5208. | 2.14 | 11.4 | 1.61 | 8.6 | 2.06 | 11 | 230 | 14 |
| 900 | 1/2 | 3.97 | 0.526 | 5246 | 1.79 | 9.66 | 3.82 | 20.6 | 2.65 | 14.3 | 300 | 62 |
| 890 | 3/4 | 4.14 | 0.555 | 4973. | 1.495 | 8.0 | 5.28 | 28.2 | 2.66 | 14.3 | 361 | 87 |
| 710 | 1/4 | 3.20 | 0.586 | 4709 | 2.50 | 10.6 | 2.41 | 10.25 | 2.78 | 11.85 | 216 | 6.3 |
| 700 | 1/2 | 3.44 | 0.606 | 4554 | 2.36 | 9.9 | 4.15 | 17.4 | 2.70 | 11.3 | 262 | 33 |
| 742 | 3/4 | 3.26 | 0.575 | 4799 | 1.69 | 7.52 | 6.53 | 29.0 | 3.26 | 14.5 | 300 | 81 |
| 447 | 1/4 | 2.49 | 0.668 | 4140 | 5.0 | 13.4 | 1.38 | 3.70 | 3.0 | 8.1 | 191 | 6. |
| 447 | 1/2 | 2.47 | 0.632 | 4366 | 5.05 | 13.4 | 4.25 | 11.40 | 3.30 | 8.8 | 205 | 6. |
| 450 | 3/4 | 2.46 | 0.552 | 4999 | 4.66 | 12.5 | 6.875 | 18.5 | 3.33 | 9.0 | 213 | 22 |
| 340 | 1/4 | 1.97 | 0.530 | 5208 | 5.3 | 10.8 | 1.43 | 2.92 | 2.31 | 4.72 | 186 | 10 |
| 332 | 1/2 | 1.93 | 0.648 | 4258 | 5.71 | 11.4 | 4.85 | 9.65 | 2.33 | 4.65 | 197 | 5 |
| 343 | 3/4 | 2.35 | 0.630 | 4381 | 5.60 | 11.9 | 8.18 | 16.8 | 3.25 | 6.7 | 219 | 6 |
| 205 | 1/4 | 1.33 | - | - | 9.10 | 10.9 | 2.58 | 3.16 | 4.1 | 5.06 | 186 | 8 |
| 205 | 1/2 | 1.33 | - | - | 8.62 | 10.6 | 7.35 | 9.04 | 3.0 | 3.64 | 189 | 11 |
| 205 | 3/4 | 1.36 | - | - | 8.45 | 10.4 | 13.0 | 16.0 | 3.6 | 4.4 | 203 | 13 |

TABLE 5.

2 FEET PIPE - 200 KG PER $CM^2$.

| SPEED R.P.M. | THROTTLE. | TIME SCALE M.M.PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT.PER SEC. | INJECT. LAG MILLI-SECS. | INJECT. LAG DEGREES. | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG. DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$ | RESID. PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 913 | ¼ | 3.76 | .544 | 5072 | 2.24 | 12.3 | 1.3 | 7.125 | 1.9 | 10.425 | 288 | 58 |
| 920 | ½ | 3.70 | 0.535 | 5158 | 1.825 | 11.0 | 3.58 | 19.8 | 2.50 | 13.8 | 333 | 88 |
| 900 | ¾ | 3.98 | 0.576 | 4795 | 1.625 | 8.76 | 4.95 | 26.8 | 2.50 | 13.56 | 391 | 98 |
| 740 | ¼ | 3.58 | 0.552 | 4999 | 2.590 | 11.45 | 1.63 | 7.24 | 2.18 | 9.69 | 286 | 72 |
| 710 | ½ | 3.55 | 0.586 | 4709 | 2.73 | 11.65 | 3.47 | 14.75 | 2.44 | 10.40 | 322 | 67 |
| 730 | ¾ | 3.50 | 0.590 | 4678 | 1.93 | 8.45 | 5.93 | 26.0 | 2.55 | 11.15 | 386 | 123 |
| 452 | ¼ | 2.93 | 0.463 | 5960 | 4.75 | 12.90 | 1.635 | 4.44 | 3.07 | 8.34 | 268 | 53 |
| 427 | ½ | 2.78 | 0.635 | 4345 | 5.48 | 14.0 | 3.94 | 10.1 | 3.16 | 8.1 | 280 | 38 |
| 427 | ¾ | 2.76 | 0.627 | 4402 | 5.03 | 12.9 | 6.65 | 17.0 | 3.08 | 7.9 | 288 | 48 |
| 383 | ¼ | 2.73 | 0.61 | 4523 | 5.0 | 11.5 | 1.18 | 2.72 | 2.28 | 5.22 | 265 | 50 |
| 380 | ½ | 2.48 | 0.682 | 4047 | 6.26 | 14.25 | 4.125 | 9.41 | 3.36 | 7.66 | 266 | 25 |
| 377 | ¾ | 2.48 | 0.71 | 3887 | 5.5 | 12.4 | 7.36 | 16.6 | 3.10 | 7.0 | 274 | 44 |
| 205 | ¼ | 1.44 | - | - | 10.47 | 12.85 | 1.082 | 1.33 | 4.2 | 5.18 | 264 | 50 |
| 205 | ½ | 1.43 | - | - | 9.77 | 12.0 | 8.30 | 10.2 | 5.04 | 6.2 | 269 | 44 |
| 205 | ¾ | 1.48 | - | - | 9.30 | 11.4 | 13.04 | 17.2 | 5.45 | 6.6 | 264 | 47 |

TABLE 6

4 FEET PIPE - 100 KG PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M.PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT.PER SEC. | INJECT. LAG MILLI-SECS. | INJECT. LAG. DEGREES. | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN. DEGREES. | CUT-OFF LAG. MILLI-SECS. | CUT-OFF LAG. DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$ | RESID. PRESS. $KG/CM^2$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 850 | ¼ | 3.11 | 1.13 | 4212 | 2.18 | 11.1 | 2.08 | 10.6 | 2.5 | 12.7 | 174 | - |
| 855 | ½ | 3.07 | 0.95 | 5009 | 1.49 | 7.64 | 4.55 | 23.3 | 2.91 | 14.94 | 222 | 42 |
| 855 | ¾ | 3.09 | 0.977 | 4871 | 1.49 | 7.64 | 6.1 | 31.3 | 3.3 | 16.94 | 277 | 45 |
| 710 | ¼ | 3.1 | 1.13 | 4212 | 2.28 | 9.7 | 2.59 | 11.0 | 2.75 | 11.83 | 170 | - |
| 710 | ½ | 3.07 | 1.11 | 4288 | 2.20 | 9.36 | 4.86 | 20.7 | 3.3 | 14.06 | 220 | 34 |
| 710 | ¾ | 3.12 | 1.036 | 4596 | 1.87 | 8.0 | 6.55 | 27.9 | 3.27 | 13.95 | 256 | 45 |
| 415 | ¼ | 2.45 | 0.937 | 5080 | 3.78 | 9.425 | 3.49 | 8.6 | 3.63 | 9.025 | 156 | 8 |
| 415 | ½ | 2.45 | 0.960 | 4957 | 3.14 | 7.82 | 6.55 | 16.3 | 3.26 | 8.12 | 193 | 19 |
| 415 | ¾ | 2.47 | 0.920 | 5172 | 2.70 | 6.72 | 9.05 | 22.5 | 2.9 | 7.22 | 211 | 43 |
| 343 | ¼ | 2.07 | 1.057 | 4502 | 5.0 | 10.3 | 3.70 | 7.60 | 4.34 | 8.93 | 145 | - |
| 330 | ½ | 2.09 | 1.146 | 4153 | 5.05 | 10 | 6.6 | 13.1 | 3.60 | 7.1 | 161 | 4 |
| 330 | ¾ | 2.06 | 1.166 | 4082 | 4.05 | 8.1 | 11.11 | 22.0 | 4.0 | 8.0 | 164 | 6 |
| 205 | ¼ | 1.39 | - | - | 9.7 | 11.9 | 2.4 | 2.96 | 4.77 | 5.86 | 147 | 8 |
| 205 | ½ | 1.29 | - | - | 9.2 | 11.3 | 7.27 | 8.95 | 3.46 | 4.25 | 158 | 17 |
| 205 | ¾ | 1.27 | - | - | 7.2 | 8.85 | 13.78 | 16.9 | 3.1 | 3.75 | 160 | 20 |

TABLE 7

4 FEET PIPE - 150 KG. PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M.PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT. PER SEC. | INJECT. LAG MILLI-SECS. | INJECT. LAG. DEGREES. | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN. DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$. | RESID. PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 898 | ¼ | 3.58 | 1.016 | 4685 | 2.45 | 13.2 | 1.86 | 10.0 | 2.64 | 14.2 | 222 | 16 |
| 905 | ½ | 3.80 | 0.85 | 5600 | 2.08 | 11.3 | 3.78 | 20.55 | 2.90 | 15.85 | 302 | 72 |
| 900 | ¾ | 3.87 | 0.942 | 5052 | 1.97 | 10.6 | 4.84 | 26.1 | 2.72 | 14.7 | 341 | 91 |
| 710 | ¼ | 3.24 | 1.027 | 4633 | 2.84 | 12.1 | 2.16 | 9.0 | 2.84 | 12.1 | 216 | 41 |
| 710 | ½ | 3.21 | 1.07 | 4448 | 2.54 | 10.7 | 3.93 | 16.75 | 2.92 | 12.44 | 273 | 52 |
| 710 | ¾ | 3.30 | 0.948 | 5020 | 2.47 | 10.52 | 5.3 | 22.50 | 3.5 | 14.7 | 290 | 75 |
| 453 | ¼ | 2.8 | 1.00 | 4760 | 5.0 | 13.55 | 2.0 | 5.4 | 3.7 | 10.1 | 191 | 31 |
| 457 | ½ | 2.85 | 0.986 | 4829 | 5.27 | 14.4 | 4.25 | 11.6 | 3.8 | 10.6 | 231 | 34 |
| 452 | ¾ | 2.84 | 1.09 | 4367 | 5.04 | 13.6 | 7.06 | 19.1 | 3.95 | 10.7 | 233 | 55 |
| 333 | ¼ | 2.12 | 1.125 | 4230 | 6.0 | 11.6 | 1.52 | 3.03 | 2.8 | 5.6 | 180 | 36 |
| 328 | ½ | 1.98 | 1.00 | 4760 | 6.34 | 12.5 | 4.21 | 8.3 | 3.3 | 6.5 | 209 | 117 |
| 328 | ¾ | 1.96 | 0.956 | 4978 | 6.13 | 12.11 | 7.34 | 14.4 | 3.05 | 6.0 | 206 | 20 |
| 205 | ¼ | 1.41 | 1.11 | 4288 | 9.675 | 11.9 | 1.55 | 1.9 | 3.9 | 4.8 | 189 | 39 |
| 205 | ½ | 1.37 | 1.114 | 4272 | 7.60 | 9.35 | 9.10 | 11.2 | 3.45 | 4.24 | 206 | 52 |
| 205 | ¾ | 1.39 | 1.041 | 4571 | 10.25 | 12.6 | 10.95 | 13.5 | 3.34 | 4.1 | 200 | 27 |

TABLE 8

4 FEET PIPE - 200 KG PER $CM^2$

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M. PER MILLI-SEC. | LAG MILLI-SECS. | WAVE VELOCITY FT. PER SEC. | INJECT. LAG. MILLI-SECS. | INJECT. LAG DEGREES | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG. DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$. | RESID. PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 898 | ¼ | 4.11 | 0.94 | 5063 | 2.62 | 14.1 | 1.495 | 8.06 | 2.44 | 13.16 | 270 | 58 |
| 948 | ½ | 4.40 | 1.04 | 4576 | 2.25 | 12.8 | 3.41 | 19.4 | 2.80 | 16.2 | 361 | 106 |
| 913 | ¾ | 4.41 | 1.01 | 4712 | 1.89 | 10.36 | 4.96 | 27.2 | 2.84 | 15.56 | 394 | 119 |
| 710 | ¼ | 3.36 | 1.05 | 4533 | 3.05 | 13.0 | 1.66 | 7.07 | 2.60 | 11.07 | 264 | 98 |
| 750 | ½ | 3.31 | 1.067 | 4463 | 3.16 | 14.2 | 3.40 | 15.3 | 3.0 | 13.5 | 337 | 92 |
| 417 | ¼ | 2.53 | 1.03 | 4620 | 5.2 | 13.0 | 1.875 | 4.6 | 3.44 | 8.61 | 270 | 86 |
| 417 | ½ | 2.50 | 1.00 | 4760 | 5.90 | 14.75 | 4.175 | 10.45 | 3.67 | 9.2 | 297 | 70 |
| 417 | ¾ | 2.48 | 0.966 | 4926 | 5.86 | 14.6 | 6.73 | 16.8 | 3.36 | 8.4 | 315 | 80 |
| 368 | ¼ | 2.48 | 0.953 | 4994 | 5.68 | 12.5 | 1.175 | 2.6 | 2.76 | 6.1 | 256 | 72 |
| 368 | ½ | 2.40 | 0.953 | 4994 | 6.70 | 14.4 | 4.21 | 9.3 | 3.48 | 7.71 | 292 | 56 |
| 368 | ¾ | 2.43 | 0.91 | 5236 | 6.18 | 13.65 | 7.04 | 15.5 | 3.24 | 7.15 | 286 | 66 |
| 205 | ¼ | 1.5 | - | - | 9.65 | 11.87 | 1.73 | 2.13 | 4.0 | 5.0 | 252 | 100 |
| 205 | ½ | 1.31 | - | - | 9.22 | 11.34 | 6.91 | 8.5 | 3.12 | 3.84 | 270 | 97 |
| 205 | ¾ | 1.24 | - | - | 9.16 | 11.26 | 11.04 | 13.57 | 2.3 | 2.83 | 270 | 95 |

TABLE 9.

8 FEET PIPE - 100 KG. PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M.PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT.PER SEC. | INJECT. LAG. MILLI-SECS. | INJECT. LAG. DEGREES | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$. | RESIDUAL PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 900 | ¼ | 3.4 | 2.35 | 3740 | 3.44 | 18.5 | 2.08 | 11.2 | 3.84 | 20.7 | 173.3 | 25 |
| 913 | ½ | 3.46 | 1.84 | 4782 | 3.08 | 16.85 | 3.83 | 21 | 4.0 | 21.85 | 247 | 36 |
| 913 | ¾ | 3.44 | 1.85 | 4756 | 2.42 | 13.25 | 5.425 | 29.7 | 3.83 | 20.95 | 298.3 | 44 |
| 674 | ¼, | 2.83 | 2.35 | 3740 | 3.68 | 14.9 | 2.135 | 8.63 | 3.60 | 14.53 | 167.1 | - |
| 674 | ½ | 2.61 | 1.99 | 4422 | 3.59 | 14.5 | 4.71 | 19.0 | 4.33 | 17.5 | 203.1 | 27 |
| 670 | ¾ | 2.66 | 1.92 | 4582 | 3.17 | 12.75 | 6.23 | 25.0 | 3.92 | 15.75 | 268.7 | 58 |
| 417 | ¼ | 2.6 | 2.27 | 3876 | 4.78 | 11.95 | 2.92 | 7.3 | 4.1 | 10.25 | 162.5 | - |
| 413 | ½ | 2.22 | 2.06 | 4303 | 4.55 | 11.3 | 6.63 | 16.4 | 4.72 | 11.7 | 190.6 | 19 |
| 417 | ¾ | 2.65 | 1.96 | 4480 | 3.78 | 9.46 | 9.04 | 22.6 | 4.0 | 10.06 | 220.2 | 47 |
| 327 | ¼ | 2.02 | 2.20 | 3999 | 5.60 | 11.0 | 3.69 | 7.23 | 4.72 | 9.23 | 156.2 | 25 |
| 380 | ½ | 2.40 | 1.95 | 4513 | 5.73 | 13.1 | 6.21 | 14.2 | 4.96 | 11.3 | 179.6 | 13 |
| 382 | ¾ | 2.38 | 2.09 | 4210 | 5.2 | 11.96 | 9.12 | 20.9 | 5.24 | 12.0 | 94.4 | 13 |
| 205 | ¼ | 1.44 | - | - | 9.85 | 12.30 | 2.535 | 3.115 | 5.20 | 6.4 | 159.3 | 30 |
| 205 | ½ | 1.41 | - | - | 11.0 | 13.5 | 6.50 | 7.995 | 4.5 | 5.5 | 156.2 | 13 |
| 205 | ¾ | 1.075 | - | - | 9.25 | 11.4 | 13.4 | 16.5 | 4.8 | 5.9 | 165.6 | 33 |

TABLE 10.

8 FEET PIPE - 150 KG PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M. PER MILLI-SECS. | LAG MILLI-SECS. | WAVE VELOCITY FT. PER SEC. | INJECT. LAG. MILLI-SECS | INJECT. LAG DEGREES. | NEEDLE OPEN. MILLI-SECS. | NEEDLE OPEN. DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG DEGREES. | MAX. NOZZLE PRESS. KG/$CM^2$. | RESID. PRESS. KG/$CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 855 | ¼ | 3.56 | 2.14 | 4111 | 3.49 | 17.90, | 2.28 | 11.70 | 4.0 | 20.6 | 198.4 | 10 |
| 882 | ½ | 3.53 | 1.85 | 4756 | 3.20 | 16.90 | 3.75 | 19.85 | 4.0 | 21.2 | 287.5 | 64 |
| 910 | ¾ | 3.57 | 1.88 | 4669 | 2.95 | 16.10 | 5.23 | 28.60 | 4.16 | 22.7 | 350 | 78 |
| 787 | ¼ | 3.64 | 1.97 | 4467 | 4.0 | 18.8 | 2.0 | 9.40 | 4.1 | 19.4 | 206.3 | 6.25 |
| 785 | ½ | 2.98 | 2.00 | 4400 | 3.70 | 18.70 | 3.68 | 17.30 | 4.24 | 20 | 258 | 57.8 |
| 795 | ¾ | 2.91 | 1.9 | 4631 | 3.34 | 15.9 | 5.80 | 27.60 | 4.5 | 21.5 | 317 | 75 |
| 447 | ¼ | 2.45 | 2.04 | 4313 | 4.80 | 12.90 | 2.80 | 7.50 | 4.25 | 11.4 | 192.1 | 47 |
| 436 | ½ | 2.45 | 1.9 | 4631 | 4.80 | 12.60 | 5.875 | 15.37 | 4.6 | 12.0 | 218.7 | 47 |
| 452 | ¾ | 2.43 | 1.875 | 4692 | 4.50 | 12.25 | 8.50 | 23.05 | 4.83 | 13.0 | 250 | 63 |
| 350 | ¼ | 2.3 | 1.95 | 4513 | 5.075 | 10.65 | 2.7 | 5.64 | 3.5 | 7.35 | 206.1 | 61 |
| 343 | ½ | 2.15 | 2.12 | 4151 | 6.20 | 12.75 | 5.82 | 12.00 | 4.25 | 8.75 | 218.7 | 42 |
| 330 | ¾ | 2.02 | 2.11 | 4170 | 7.0 | 13.06 | 8.025 | 15.84 | 4.66 | 9.24 | 236 | 40 |
| 205 | ¼ | 1.35 | 2.08 | 4230 | 9.9 | 11.20 | 2.86 | 3.52 | 4.65 | 5.72 | 179.6 | 64 |
| 205 | ½ | 1.28 | 1.95 | 4513 | 12.1 | 14.88 | 6.675 | 8.20 | 5.76 | 7.08 | 190.6 | 30 |
| 205 | ¾ | 1.175 | - | - | 9.1 | 10.20 | 15.15 | 18.60 | 5.53 | 6.8 | 199.7 | 60 |

TABLE 11.

8 FEET PIPE - 200 KG. PER $CM^2$.

| SPEED R.P.M. | THROTTLE | TIME SCALE M.M. PER MILLI-SEC. | LAG MILLI-SECS. | WAVE VELOCITY FT. PER SEC. | INJECT. LAG MILLI-SECS. | INJECT. LAG DEGREES. | NEEDLE OPEN MILLI-SECS. | NEEDLE OPEN DEGREES. | CUT-OFF LAG MILLI-SECS. | CUT-OFF LAG DEGREES. | MAX. NOZZLE PRESS. $KG/CM^2$. | RESID. PRESS. $KG/CM^2$. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 948 | ¼ | 4.34 | 1.94 | 4631 | 4.10 | 23.30 | 1.175 | 6.70 | 3.7 | 21 | 281 | 52 |
| 948 | ½ | 3.96 | 2.10 | 4191 | 3.74 | 21.30 | 2.92 | 16.60 | 3.85 | 21.9 | 378 | 94 |
| 913 | ¾ | 3.95 | 1.95 | 4513 | 3.40 | 18.65 | 4.67 | 25.60 | 4.06 | 22.25 | 392 | 114 |
| 708 | ¼ | 3.07 | 1.93 | 5559 | 4.10 | 17.40 | 1.73 | 7.35 | 3.70 | 15.75 | 277 | 73 |
| 730 | ½ | 3.03 | 1.88 | 4680 | 3.68 | 16.10 | 4.275 | 18.70 | 4.30 | 18.8 | 290 | 81 |
| 750 | ¾ | 3.07 | 1.88 | 4680 | 3.12 | 14.00 | 5.57 | 25.15 | 3.8 | 17.15 | 381 | 120 |
| 436 | ¼ | 2.83 | 1.91 | 4607 | 4.98 | 13.00 | 2.425 | 6.34 | 3.95 | 10.34 | 281 | 106 |
| 436 | ½ | 2.88 | 1.91 | 4607 | 5.8 | 15.2 | 5.14 | 13.40 | 4.45 | 11.65 | 280 | 86 |
| 436 | ¾ | 2.55 | 2.09 | 4211 | 5.32 | 13.90 | 8.1 | 21.2 | 5.0 | 13.1 | 290 | 94 |
| 376 | ¼ | 2.19 | 1.91 | 4607 | 5.15 | 11.60 | 3.1 | 6.9 | 4.2 | 9.5 | 264 | 111 |
| 376 | ½ | 2.18 | 1.91 | 4607 | 6.42 | 14.5 | 5.3 | 11.99 | 4.65 | 10.49 | 274 | 77 |
| 376 | ¾ | 2.20 | 1.94 | 4536 | 5.75 | 13 | 8.03 | 20.4 | 5.08 | 11.45 | 284 | 61 |
| 205 | ¼ | 1.48 | 2.18 | 4037 | 9.0 | 11.07 | 3.0 | 3.97 | 4.91 | 6.04 | 244 | 103 |
| 205 | ½ | 1.36 | 2.16 | 4074 | 11.35 | 13.88 | 6.83 | 8.47 | 5.20 | 6.35 | 253 | 87 |
| 205 | ¾ | 1.18 | 2.03 | 4335 | 9.75 | 11.98 | 12.3 | 15.13 | 4.33 | 5.33 | 250 | 99 |

DISCHARGE CHARACTERISTICS in RELATION to PUMP SPEED, THROTTLE OPENING, PIPE LENGTH and NEEDLE TENSION.

(iii) (a) Initiation and Termination of Injection. From the point of view of practical operation, the important features of the injection process are the initiation and termination of injection in relation to pump and hence engine movement, the duration of the discharge and the quantity and atomisation of the injected fuel. With the exception of the latter phenomenonwhich can only be indirectly studied from the pressure and needle lift records of the present work, but which has been extensively investigated by several workers including Juhaz[25], the above mentioned features have been observed over a fairly wide range of pump speed and load operations.

Dealing firstly with the important questions of initiation, termination and duration of injection, the observed results in Tables 3-11 have been plotted as curves on a basis of pump speed. The curves are not exact but give a general indication of injection tendencies. Check records made at a few intermediate points agreed very closely with the drawn curves but the writer is not prepared to say that discontinuities do not exist.

Two series of curves have been drawn with time and degrees of pump angle as ordinates respectively. (Figs 59-85). In each series, the lower curve denotes the point of opening of the needle, the upper curve the termination of injection. Both time and pump degrees are measured from the arbitrary zero represented by the first indication of pressure rise at the pressure recorder near the pump. The release curve included in each series is also measured from the same zero as explained in Section 2, Part IV. Vertical ordinates between the needle opening and needle closing curves give the duration of injection at a given pump speed. Similarly the ordinate between the upper curve and release curve in each series denotes the cut-off lag. Comparative injection lags are measured between the base and the needle opening curve.

Before discussing the curves, it will be advisable to consider, briefly, certain operating characteristics peculiar to the injection system investigated. In contradistinction to the open nozzle injection system, wherein initial discharge is effected against a more or less constant pressure determined by engine operating conditions, the spring-loaded system encloses in the fuel-line between successive injections a volume of oil at a pressure varying for given pump and nozzle dimensions with pump speed, load and pipe length. This residual pressure exercises considerable influence upon the injection characteristics, more particularly on the injection lag, and so many factors serve to determine its magnitude that this one question would of itself form the subject for an extensive research. In the present work, it will suffice to briefly consider the observed results in the light of certain factors affecting their magnitude and to note the influence of residual pressure variation upon injection characteristics.

In the general analysis of the injection characteristics, it was noted that at the higher pump speeds, the pressure in the system increased with increasing speed throughout the discharge period, and, for a given speed, with increasing throttle. On the other hand, at the lower pump speeds due to the increasing influence of needle movements on relatively weak pressure waves considerable fluctuation in pressure persisted throughout the injection, more particularly with the longer pipe lengths and higher needle settings. Since therefore in the system under investigation, the fuel-line is unloaded by a constant volume at all speed and load conditions it might be expected that at higher pump speeds residual pressures would increase with both speed and throttle increase, would fluctuate at the lower speeds, and for all speeds and throttle settings increase with increasing needle tension.

A survey of the results tabulated in Col. 13, Tables 3-11 reveals that in general residual pressures:

(1) Increase for a given pump speed and pipe length with increasing needle tension.

(2) Increase for a given pump speed with increase in pipe length at speeds below 450 r.p.m.

(3) Increase with throttle increase at a nominal speed of 900 r.p.m. for all needle tensions investigated, and at nominal speeds of 700 and 900 r.p.m., for the lower needle tensions of 150 and 200 KG per $CM^2$.

(4) Vary irregularly under conditions other than outlined above.

(5) Vary in magnitude for a given throttle setting throughout the entire speed range investigated rising and falling as speed is increased and invariably showing relatively high values at the lowest speed examined (200 r.p.m.)

It is evident from this broad survey that while in general residual pressures tend to conform to the simple premises outlined, there are nevertheless certain features which do not admit of such ready explanation. In seeking to account for these, it becomes necessary to consider certain factors which may influence the magnitude of the residual pressures.

In the first place, the mechanism of needle closing may be expected to exert a considerable influence on residual pressure values as determining the duration of fuel expansion through the nozzle after release has been effected at the pump. Upon the arrival of the negative wave following pump release, the needle will close in a manner determined by three principal factors.

(1) The pressure obtaining at the needle at the moment of arrival of the negative wave.

(2) The intensity of the negative wave.

(3) The position of the needle relatively to its seat.

With regard to the first point, the present experiments have shown that at pump speeds in excess of 400 r.p.m. the pressure at the needle increases throughout the discharge with both speed and throttle increase; below this speed pressure conditions fluctuate. The maximum pressures at the needle for $\frac{1}{4}$ throttle opening vary but little throughout the speed range for a given pipe length and needle setting.

The intensity of the negative wave is dependent upon the rate of closing of the delivery valve and the velocity of the fuel in the line at release. Both factors are functions of pump plunger velocity since the closing of the delivery valve is dependent upon the rate of uncovering of the spill port.

The time of closing will also depend on the distance of the delivery valve from its seat and this again is dependent on plunger velocity increasing with increasing speed and throttle opening.

At the lower pump speeds and smallest throttle openings at higher speeds the needle may not as the records clearly show attain to its full lift, but float near its seat. Consequently the arrival of even relatively weak negative waves may cause the needle to seat comparatively quickly thus sealing the line at a relatively high pressure. On the other hand when at the higher speeds and throttle settings, the needle is raised to its full lift and maintained in that position throughout the injection period, needle closure may be relatively delayed on account of the closing inertia even after the arrival of a comparatively strong negative wave. In this manner, the fuel may continue to expand through the nozzle to a pressure below the normal closing pressure of the needle.

Closely associated with (2) above, as affecting the expansion of the fuel following release is the nature of the release action. In contra distinction to the opening of the needle which takes place largely as the result of pressure wave propagation, the termination of injection is dependent upon hydrostatic action to a degree varying with pump speed. This effect is particularly noticeable at the lower speeds where owing to the relatively slow uncovering of spill port, the fuel expands slowly to atmospheric pressure befôre the unloading action comes into effect. As was shown in Part IV Section(iii)the effect of the unloading valve in determining the rate of needle closing is comparatively small at the lower pump speeds. Owing to the greater fluid volume and fluid resistance, it is reasonable to suppose that with increasing pipe length, the pressure drop due to hydrostatic action (speed and loading conditions remaining constant) will be less and the residual pressure correspondingly greater, the effect increasing with increasing needle tension.

Finally, leakage through either the delivery valve or needle during the interval between successive injections introduces a disturbing factor largely dependent upon the mechanical condition of the parts but not independent of operating conditions as shown by Le Mesurier and Stansfield (11).

Examination of these several factors shows that residual pressures are not built up in a regular manner but dependent as they are upon the resultant effect of several factors varying in magnitude with load and speed change may be expected to show considerable variation throughout the speed range of the pump.

It is seen that owing to effects associated with needle closing in combination with hydrostatic expansion, pressures at the lower speeds are of greater magnitude than simple consideration of line-pressure and unloading effect would lead one to expect. This feature is particularly noticeable with the longer pipe lengths at the higher needle and lower throttle settings.

Returning to the question of the discharge characteristics, the important influence of the residual pressure is at once evident from a study of the figures and curves relating to injection lag. Referring to the figures in Col. 6 Tables 3-11 and to the corresponding curves in Figs. 59-85, the following general statements relating to needle-opening characteristics may be made:

(1) The Injection lag INCREASES in TIME.

(a) With decreasing pump speed under constant throttle, needle tension and pipe-length conditions.

(b) With increasing needle tension, regularly with higher throttle settings at higher speeds, irregularly at lower speeds and low throttle settings at the higher speeds.

(c) With increasing pipe length at the higher speeds for constant speed, throttle and needle tension.

(2) The injection lag decreases relatively with increasing pipe length at the lower speeds with increasing needle tension, other conditions remaining constant, and tends towards a constant

value independent of pipe length.

(3) The injection lag DECREASES in time with increasing throttle at the higher pump speeds for constant speed needle tension and pipe length.

(4) The injection lag varies irregularly in time under conditions other than those enumerated above.

(5) For a given pump speed, needle tension and pipe length, the lag with varying throttle varies with residual pressure, decreasing with increasing residual pressure.

(6) The irregular opening characteristics noted in (4) above may be directly correlated with corresponding variations in residual pressure, higher values of injection lag occurring with lower residual pressures.

(7) Expressed in DEGREES of pump angle (vide Col. 7, Tables 3-II and curves Figs. 59-85,), the injection lag decreases at the higher speeds with increasing throttle for constant speed, needle tension and pipe-length.

(8) In DEGREES the injection lag generally increases with increasing needle tension, but is irregular at the lowest speed examined. (200 R.P.M.).

(9) In DEGREES the injection lag varies with speed in a manner dictated largely by the length of pipe as affecting the residual pressure. As may be observed from the needle opening curves in the upper series of Figs.59-85 the injection lag for the two feet pipe tends towards a falling characteristic with increase in speed, that for the four feet pipe a constant or slightly rising characteristic while the eight feet pipe shows an almost continuous increase. The characteristic for all pipe lengths shows a relatively small lag at the lowest speeds examined.

The outstanding feature of the injection lag characteristic is the close relationship existing between lag and residual pressure, and the marked influence of the latter at the lower speeds with higher needle tensions and with longer pipes. So great does this become that injection lag is in no way comparable with pipe length.

That the injection lag should increase in time with degrease in speed is only to be expected in view of the decreasing rate of building up pressure in the fuel line. Owing however to the factors introduced by varying residual pressures, the time variation is not a simple function of the speed and the corresponding lag expressed in degrees of pump angle shows considerable fluctuation throughout the load and speed range.

In contradistinction to the injection lag which in general shows a marked increase in time with decreasing pump speed, the cut-off lag while showing a tendency to increase in time with decrease in speed remains relatively unaffected by speed variation. Accordingly, as may be observed from Figs. 59-85, there is a fairly regular increase in the termination of injection with increasing pump speed when measured in degrees of pump angle.

In general, the cut-off lag increases both in time and degrees with increasing pipe length, but shows no direct relation to increased needle tension for constant pipe-length.

It would appear that the cut-off lag is largely dependent upon the rate of closing of delivery-valve and needle. At the higher speeds owing to the rapid uncovering of the spill port, the delivery valve seats relatively quickly thus setting up a comparatively strong negative wave. This is, however associated with high pressures at the nozzle, and full needle lift thus making the time of termination of injection at the higher speeds comparable with those existing at the lower speeds, where a weaker negative wave is largely offset by the near approach of the needle to its seat and lower pressures at the nozzle.

FIGS. 59 - 85.

DURATION of DISCHARGE.

FIG. 59. 1/4 100 2 ft.

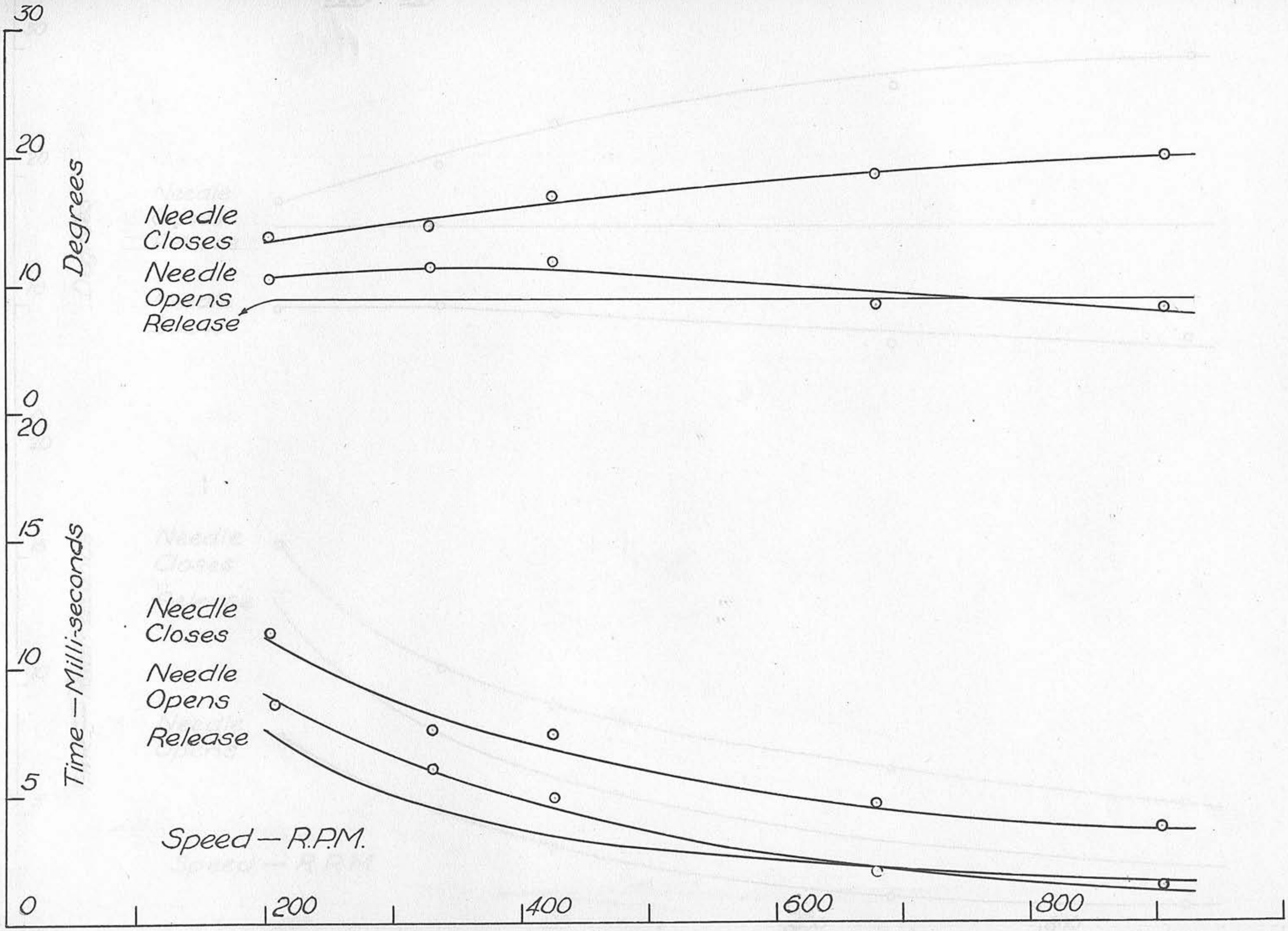


**FIG. 60.** ½ 100 2 ft.

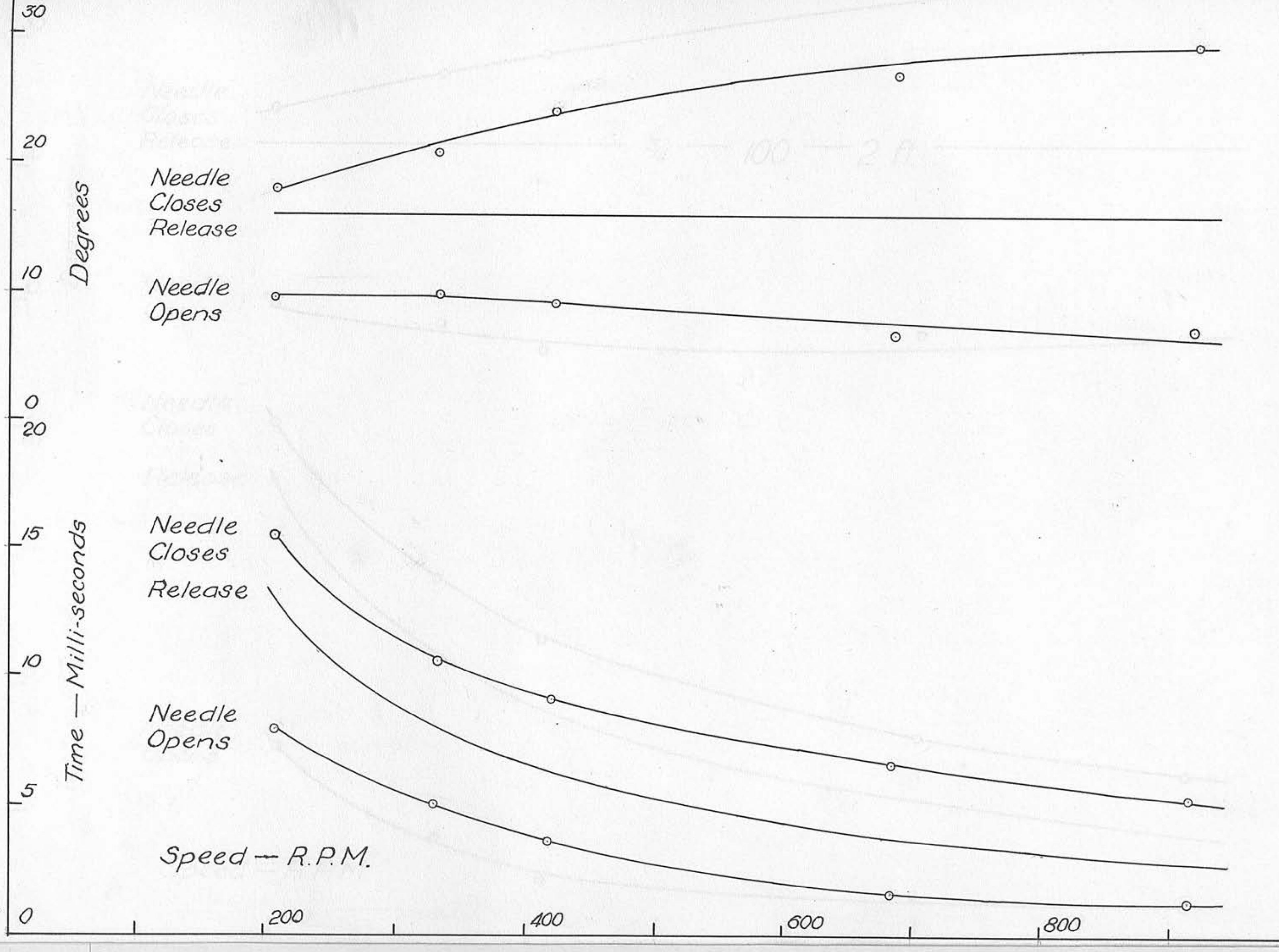
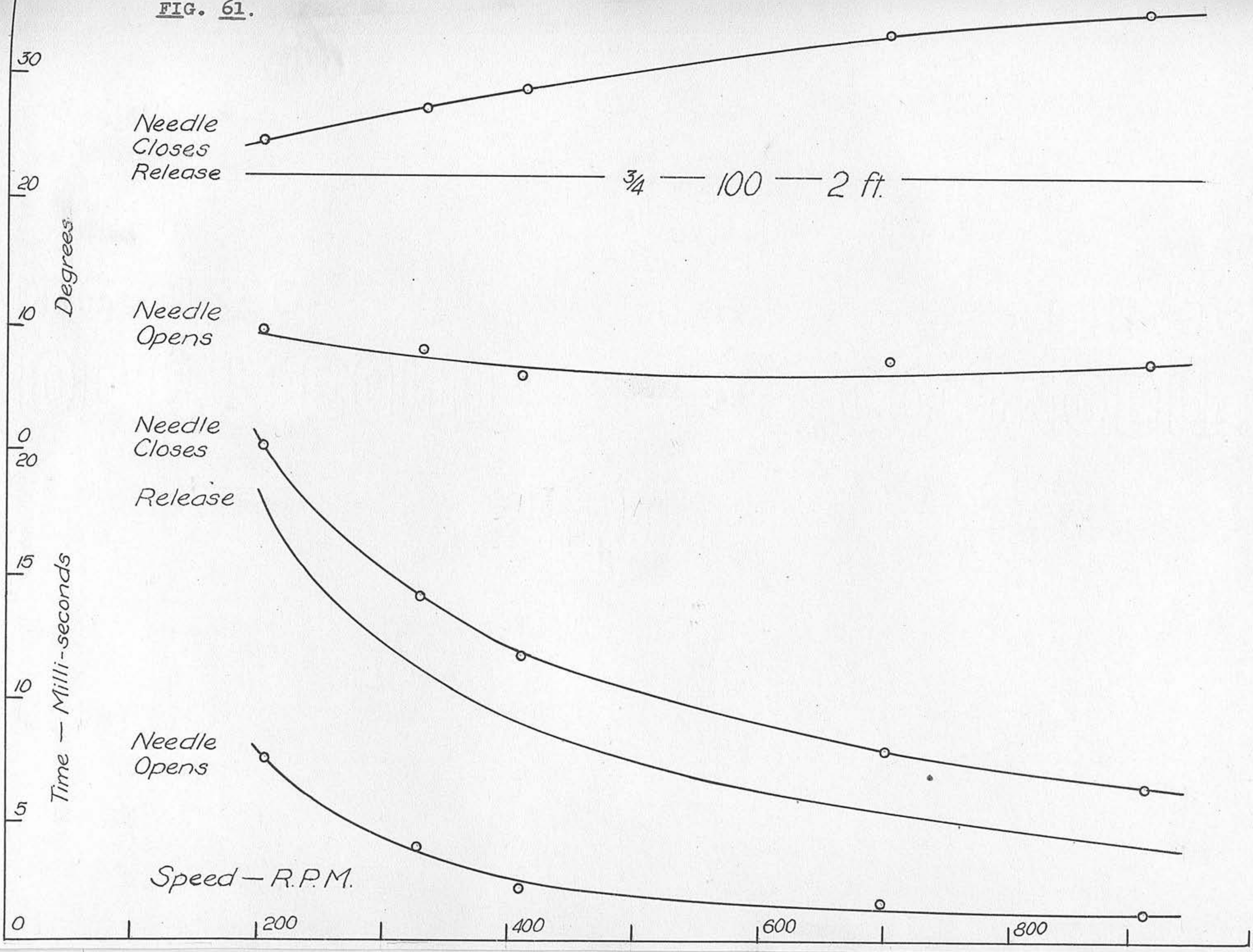


**FIG. 61.**



Needle Closes
Release
3/4 — 100 — 2 ft.
Needle Opens
Needle Closes
Release
Needle Opens
Speed — R.P.M.
Degrees
Time — Milli-seconds
30
20
10
0
20
15
10
5
0
200
400
600
800

FIG. 62. ¼ 150 2 ft.

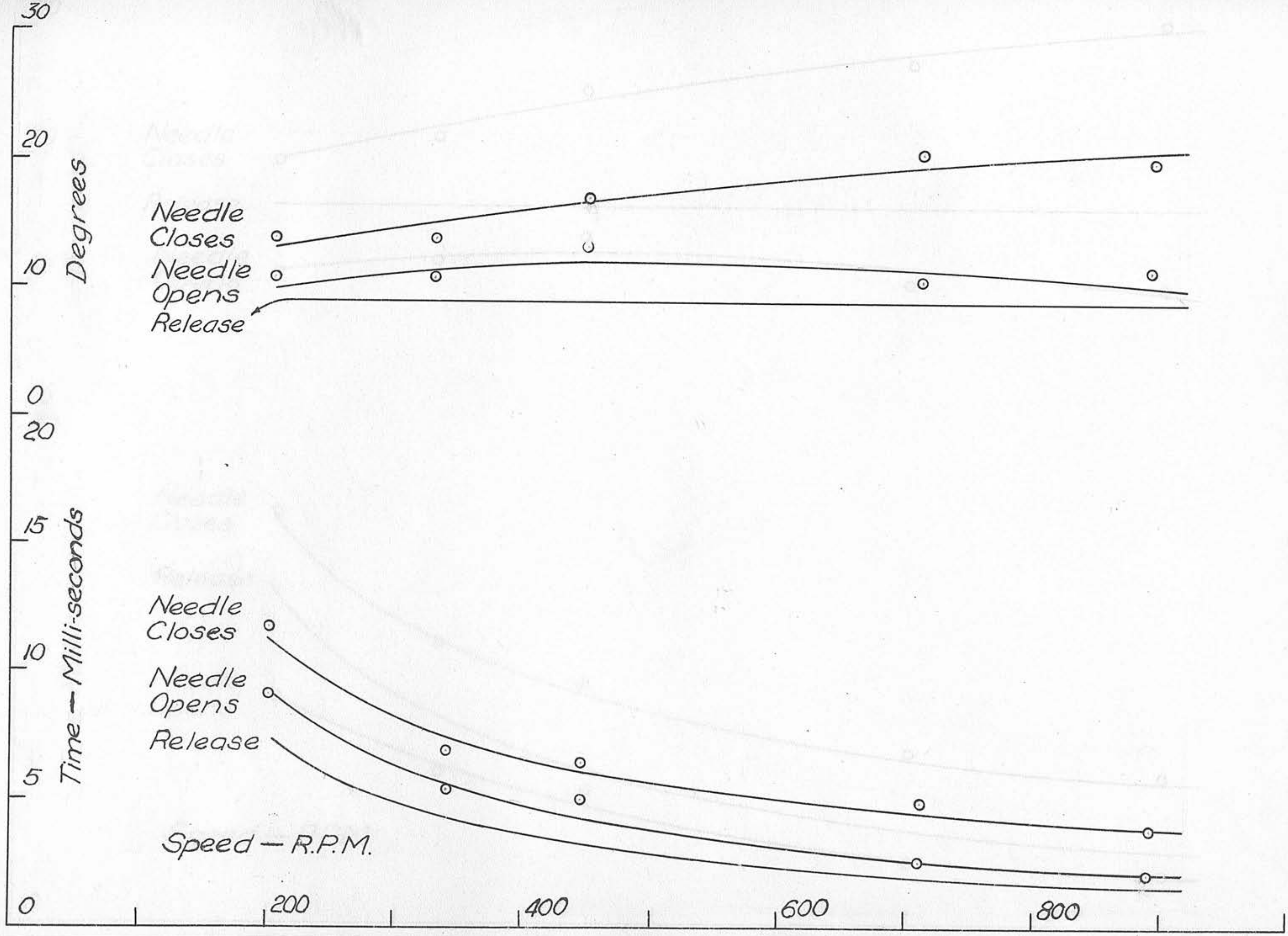


FIG. 63.

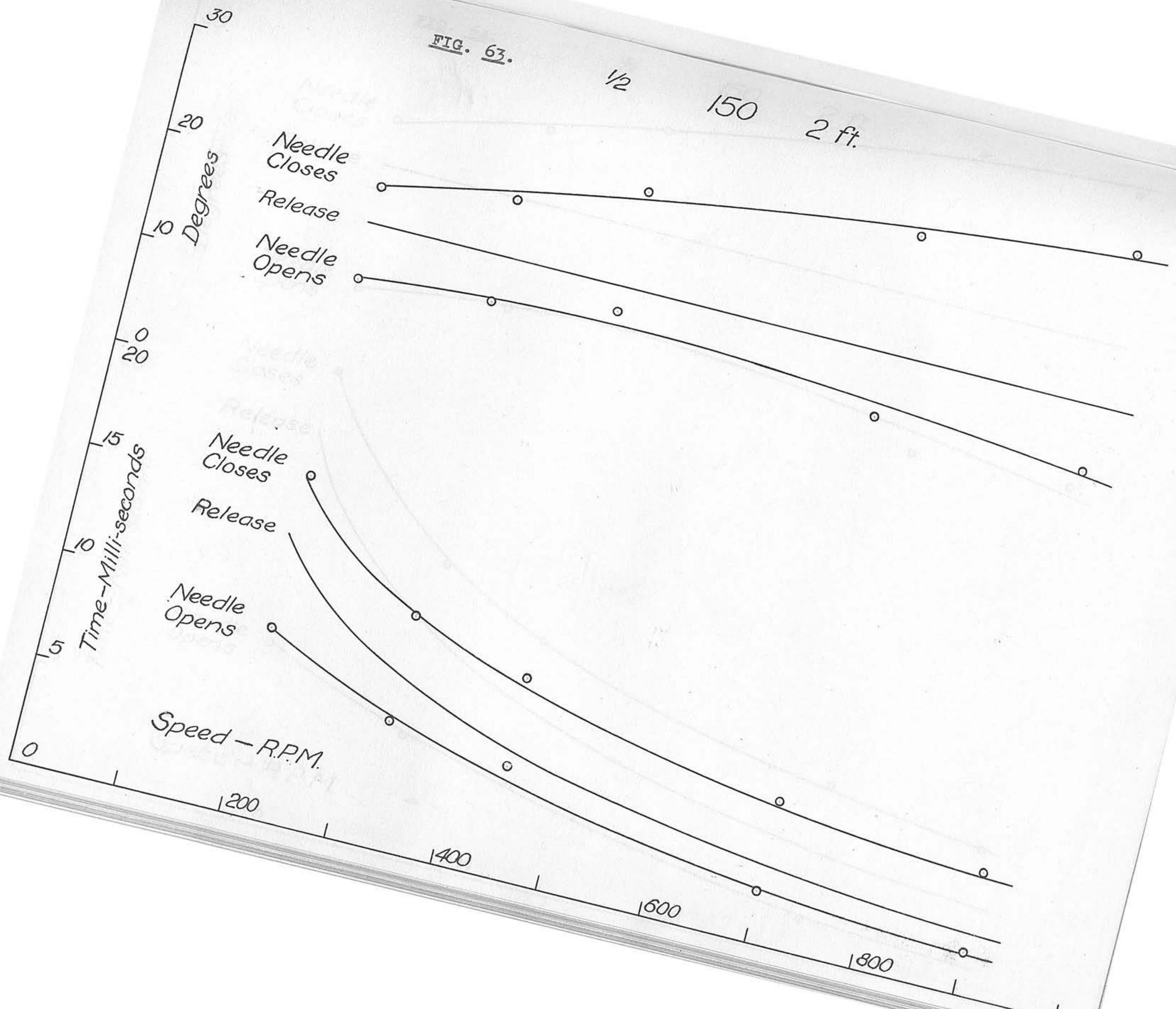

1/2
150
2 ft.
Needle Closes
Release
Needle Opens
Degrees
30
20
10
0
Needle Closes
Release
Needle Opens
Time-Milli-seconds
20
15
10
5
0
Speed - R.P.M.
200
400
600
800

**FIG. 64.** 3/4 150 2 ft.

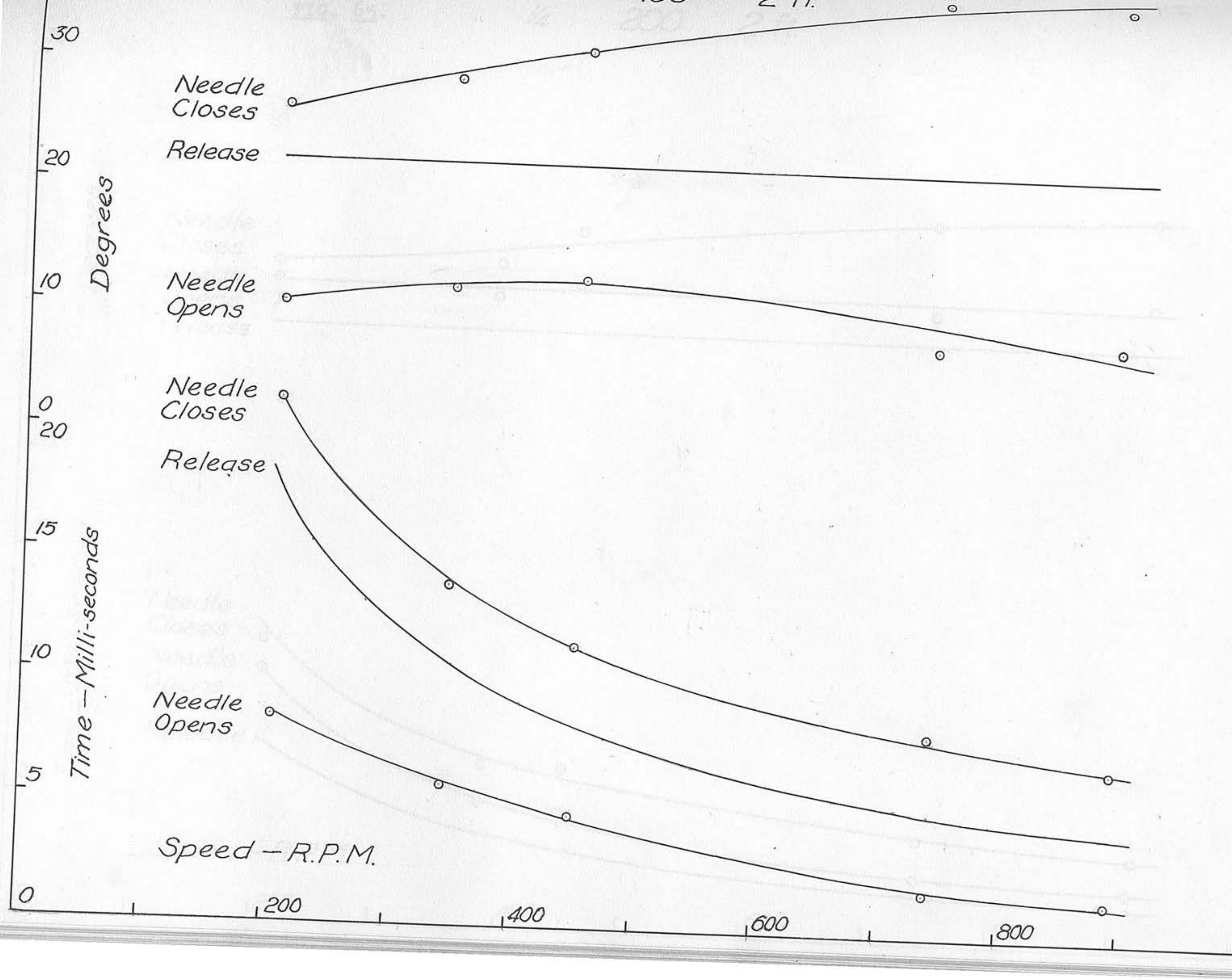


FIG. 65. ¼ 200 2 ft.

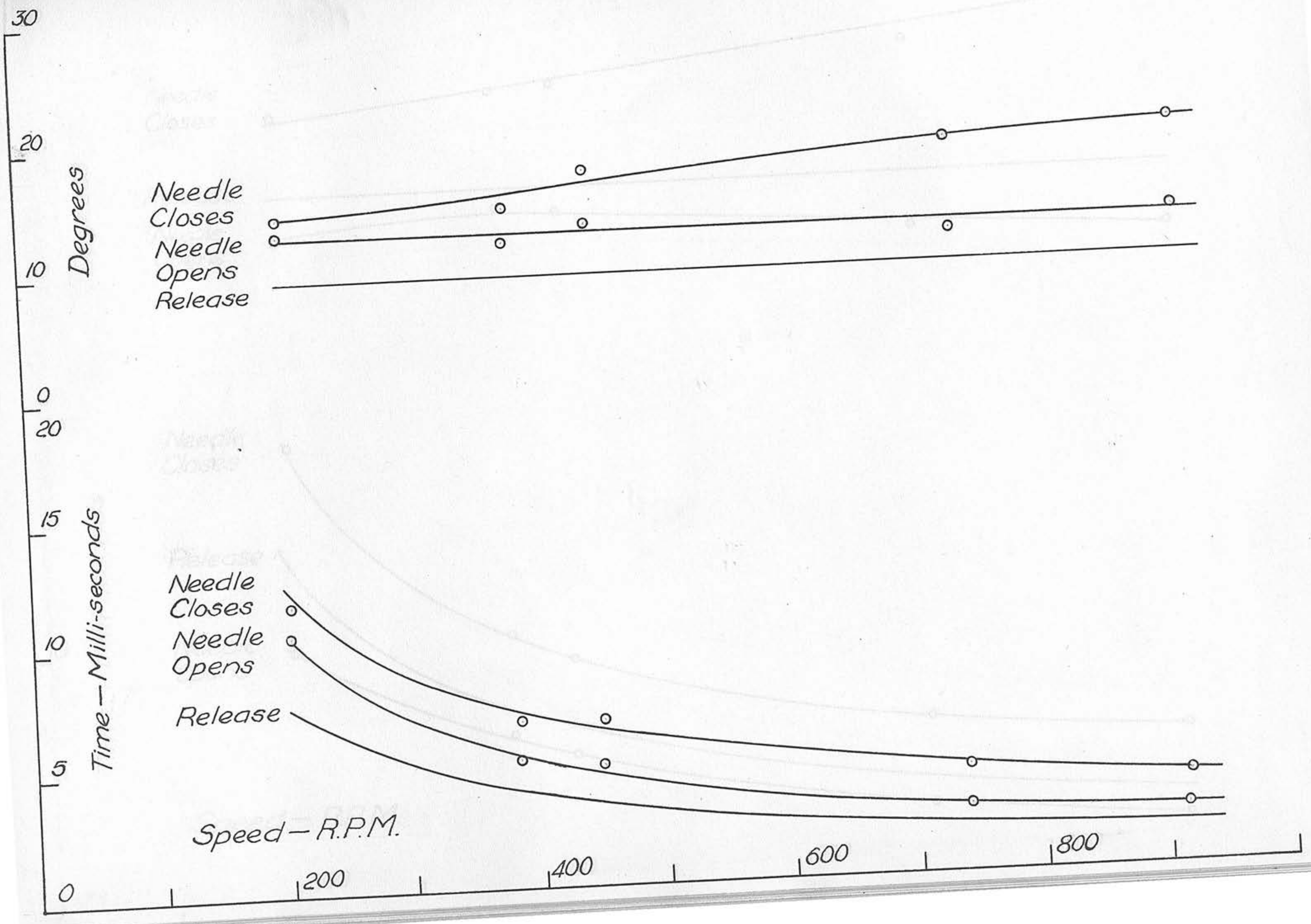


FIG. 66. ½ 200 2 ft.

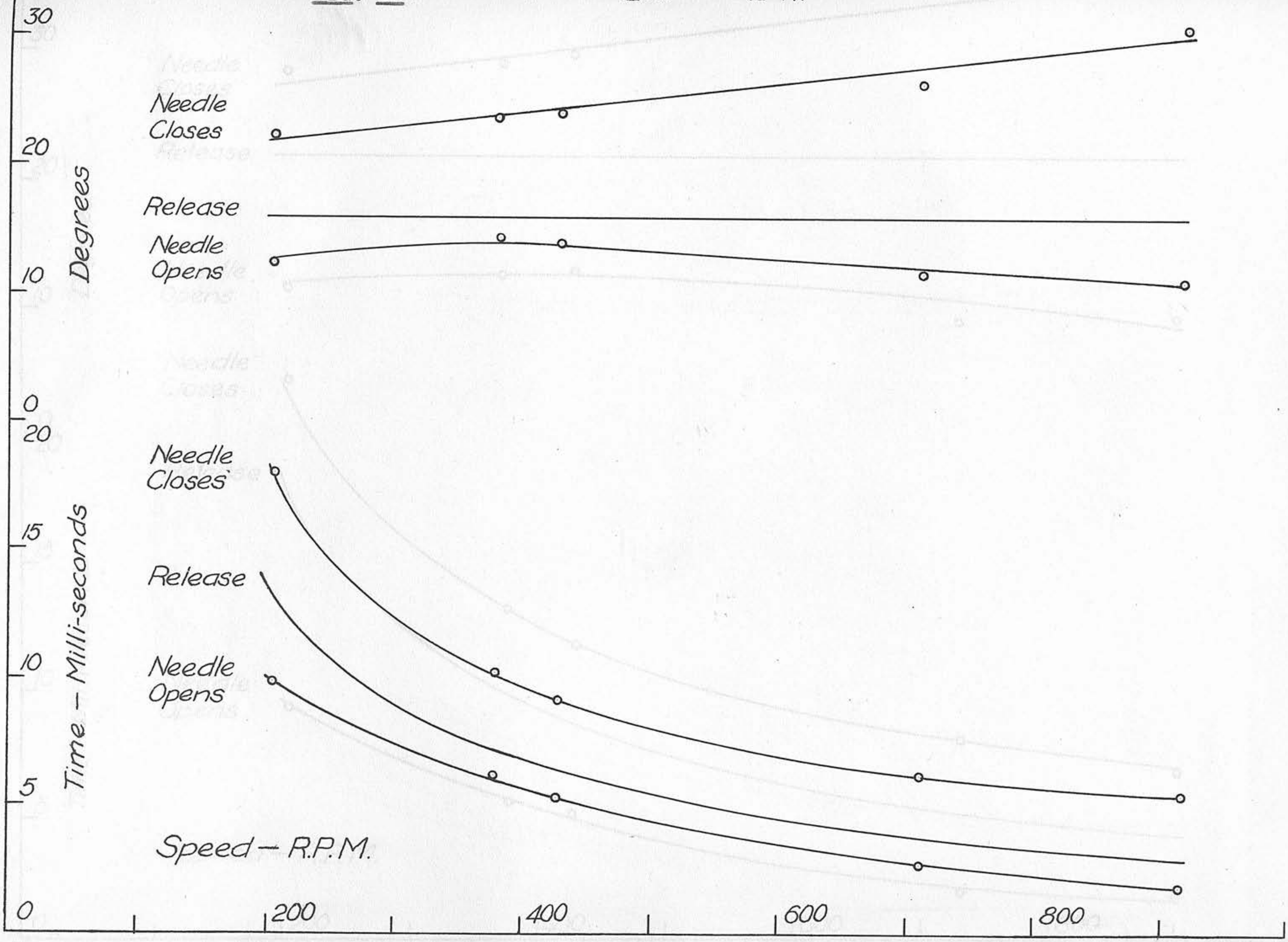
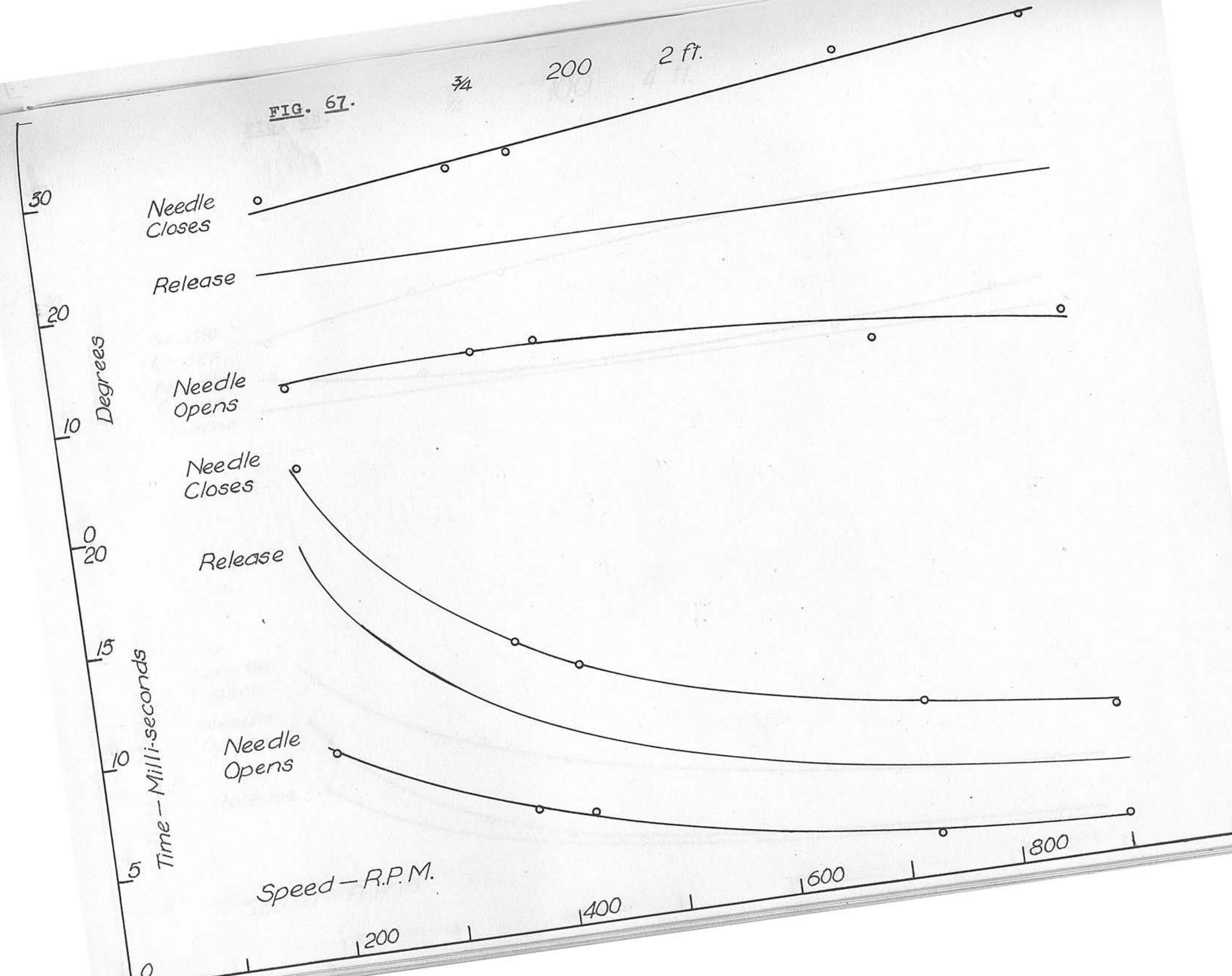


FIG. 67.



3/4
200
2 ft.
Needle Closes
Release
Needle Opens
Needle Closes
Release
Needle Opens
Degrees
30
20
10
0
20
15
10
5
0
Time – Milli-seconds
Speed – R.P.M.
200
400
600
800

FIG. 68. ¼ 100 4 ft.

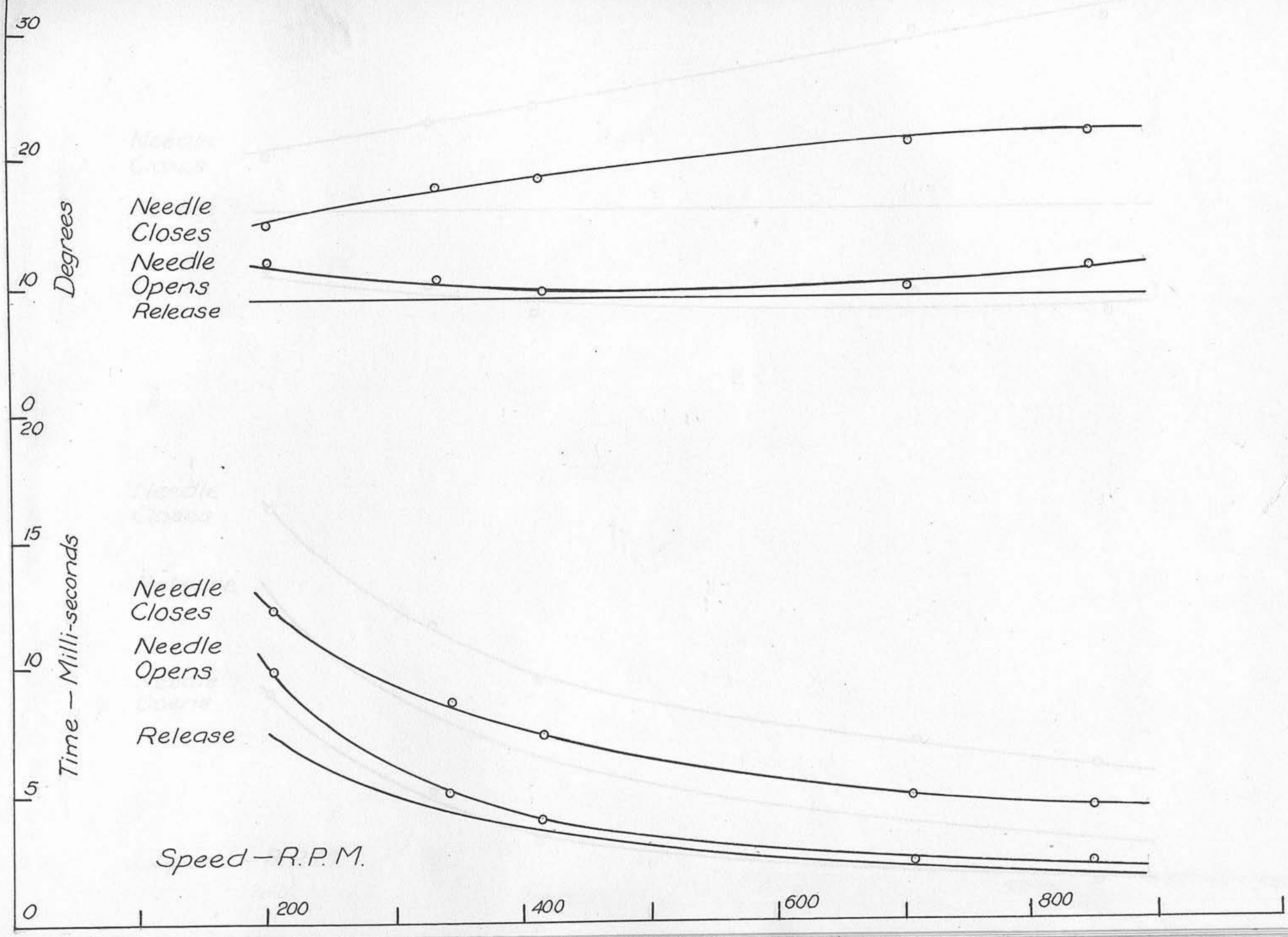


FIG. 69.    1/2    100    4 ft.

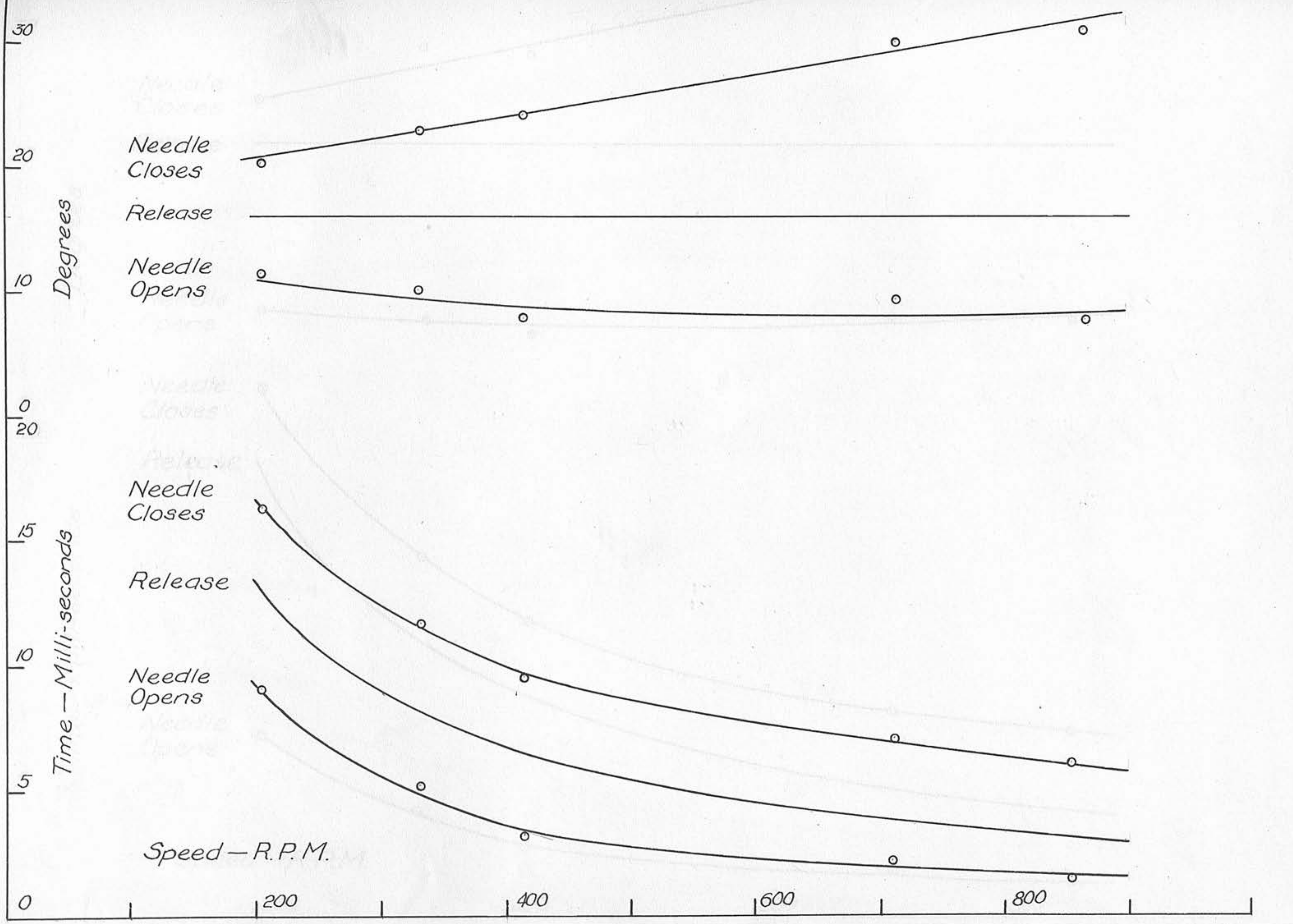
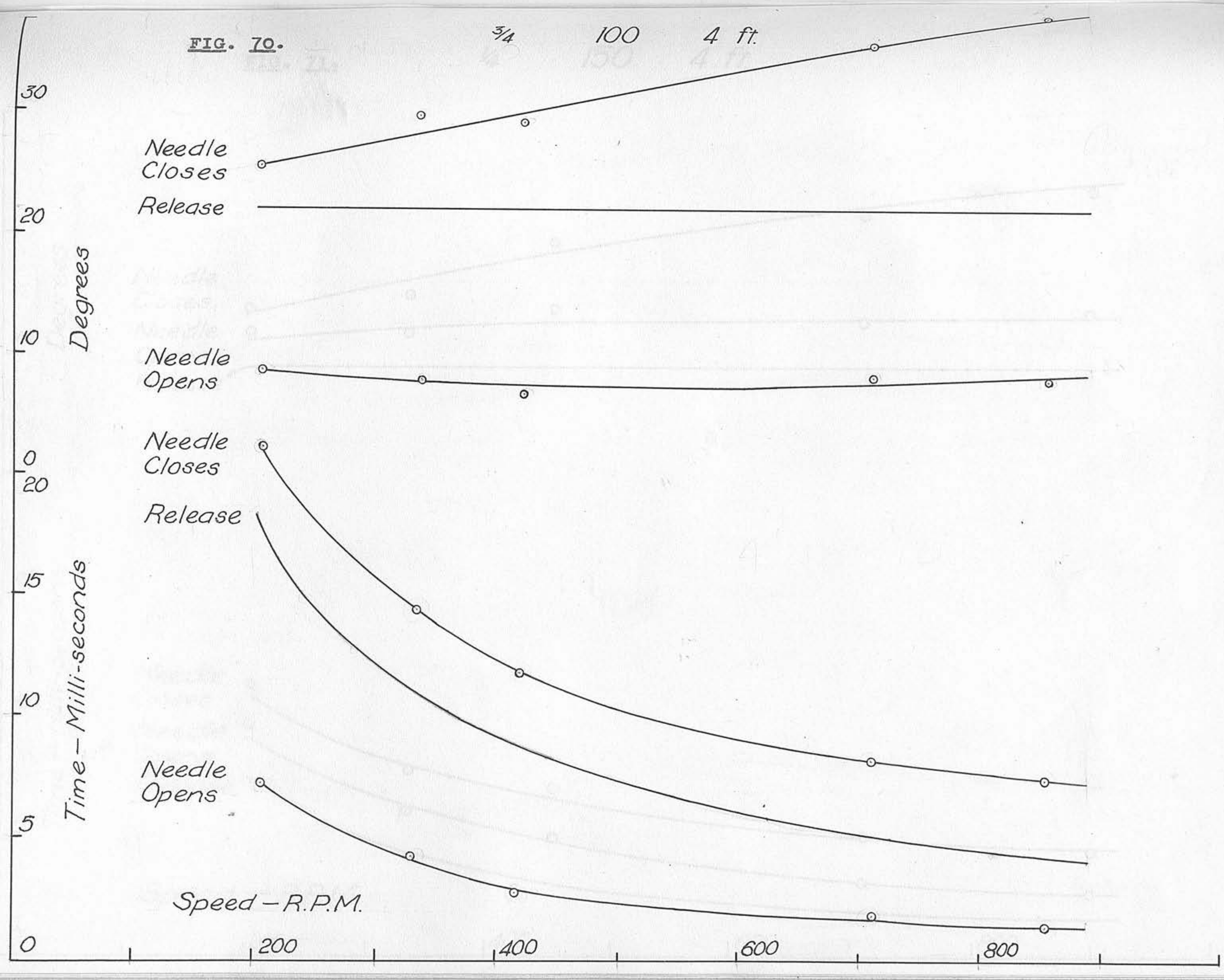


FIG. 70.



3/4 100 4 ft.
Needle Closes
Release
Needle Opens
Needle Closes
Release
Needle Opens
Degrees
Time – Milli-seconds
Speed – R.P.M.
30
20
10
0
20
15
10
5
0
200
400
600
800

**FIG. 71:** ¼ 150 4 ft.

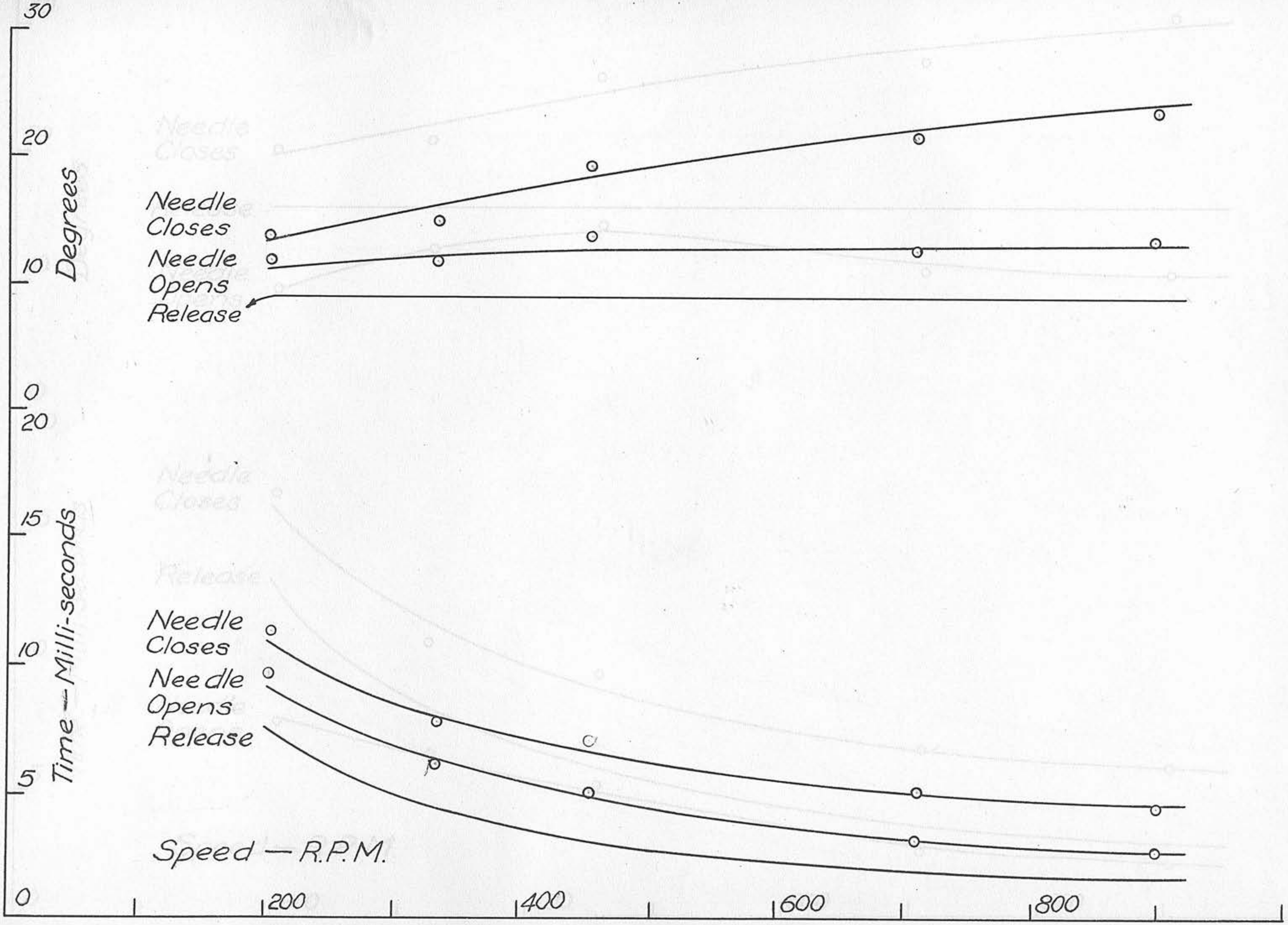


FIG. 72. ½ 150 4 ft.

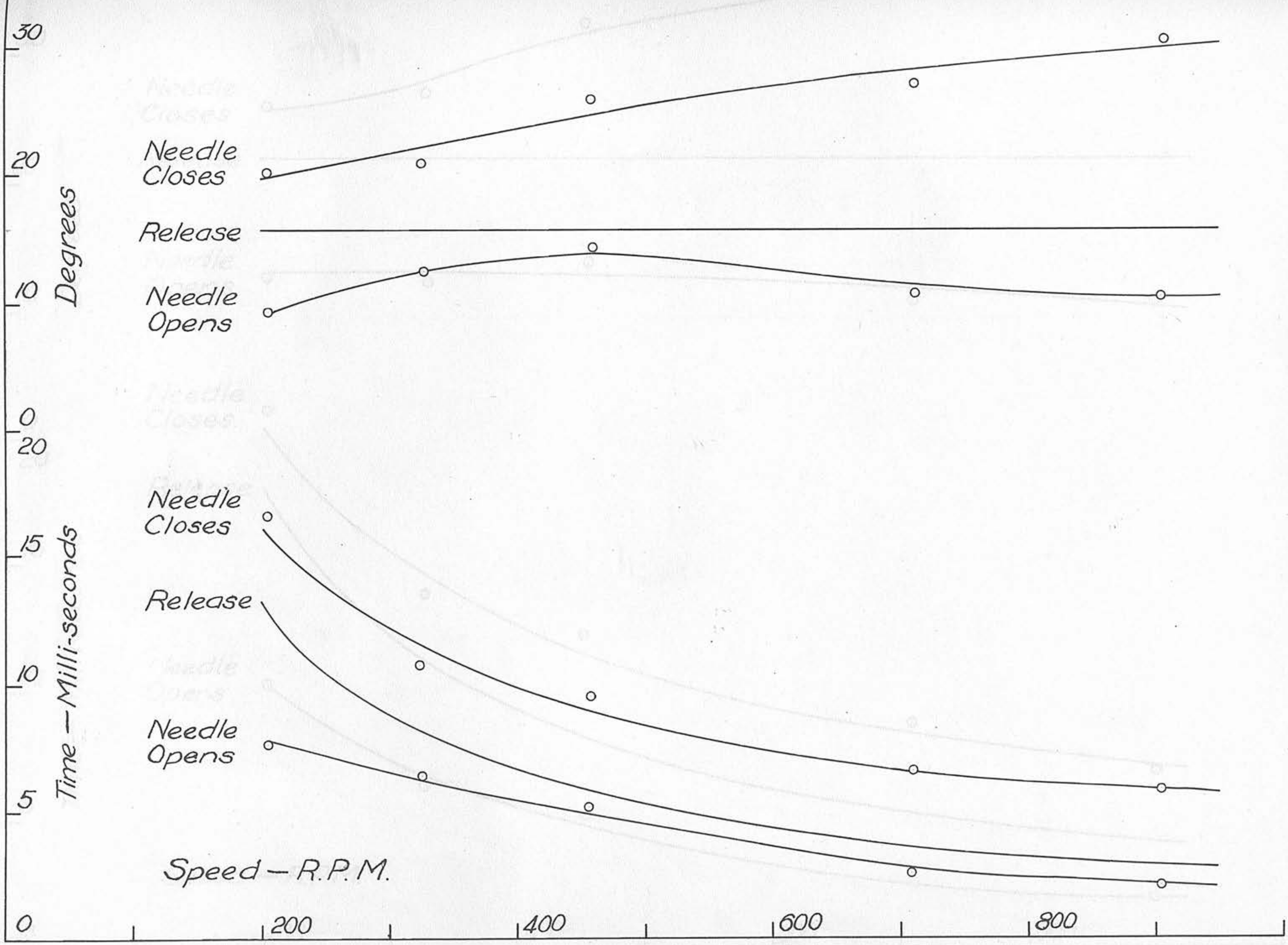


FIG. 73.

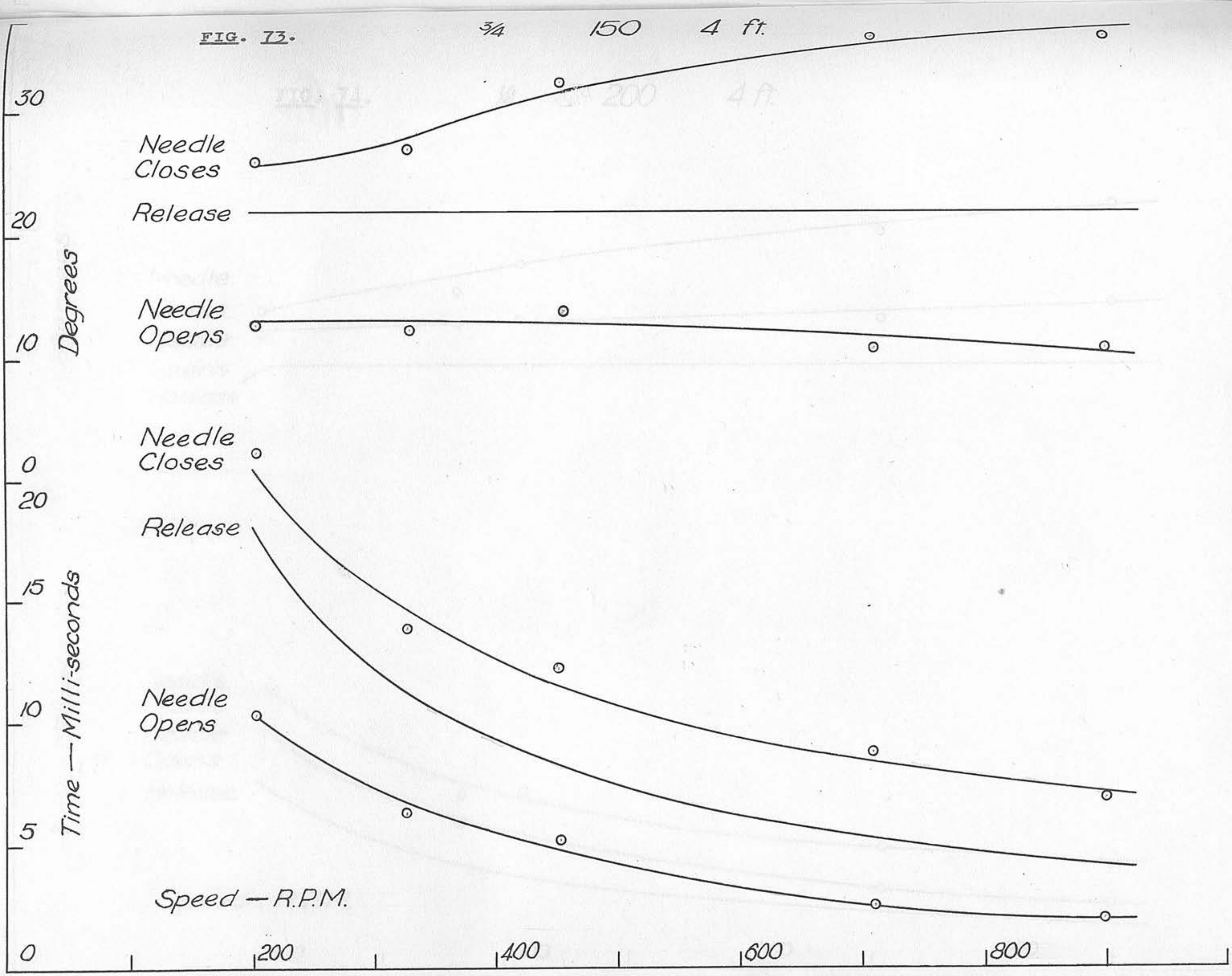

3/4 150 4 ft.
Needle Closes
Release
Needle Opens
Needle Closes
Release
Needle Opens
Degrees
Time — Milli-seconds
Speed — R.P.M.
30
20
10
0
20
15
10
5
0
200
400
600
800

FIG. 74. ¼ 200 4 ft.

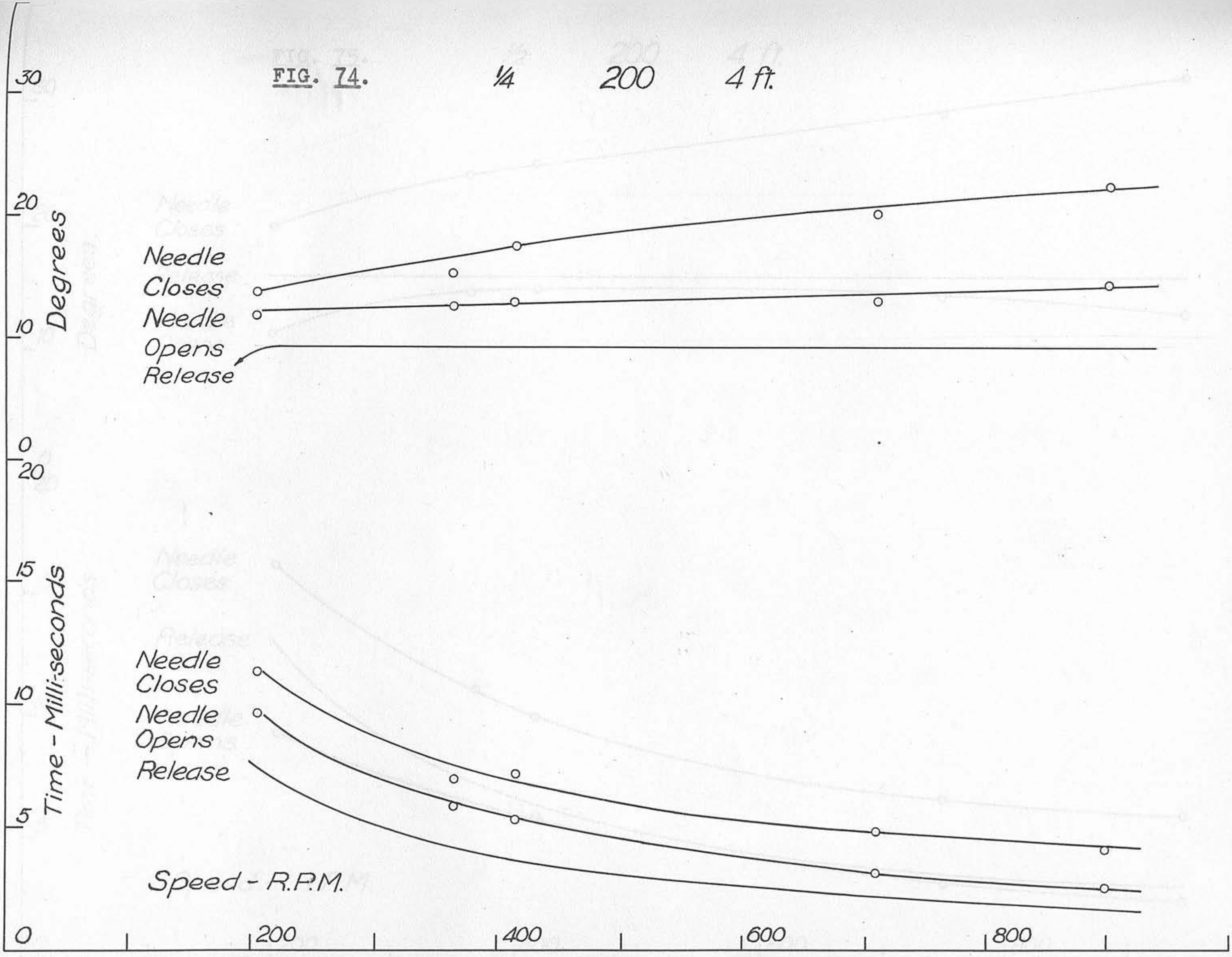


FIG. 75. ½ 200 4 ft.

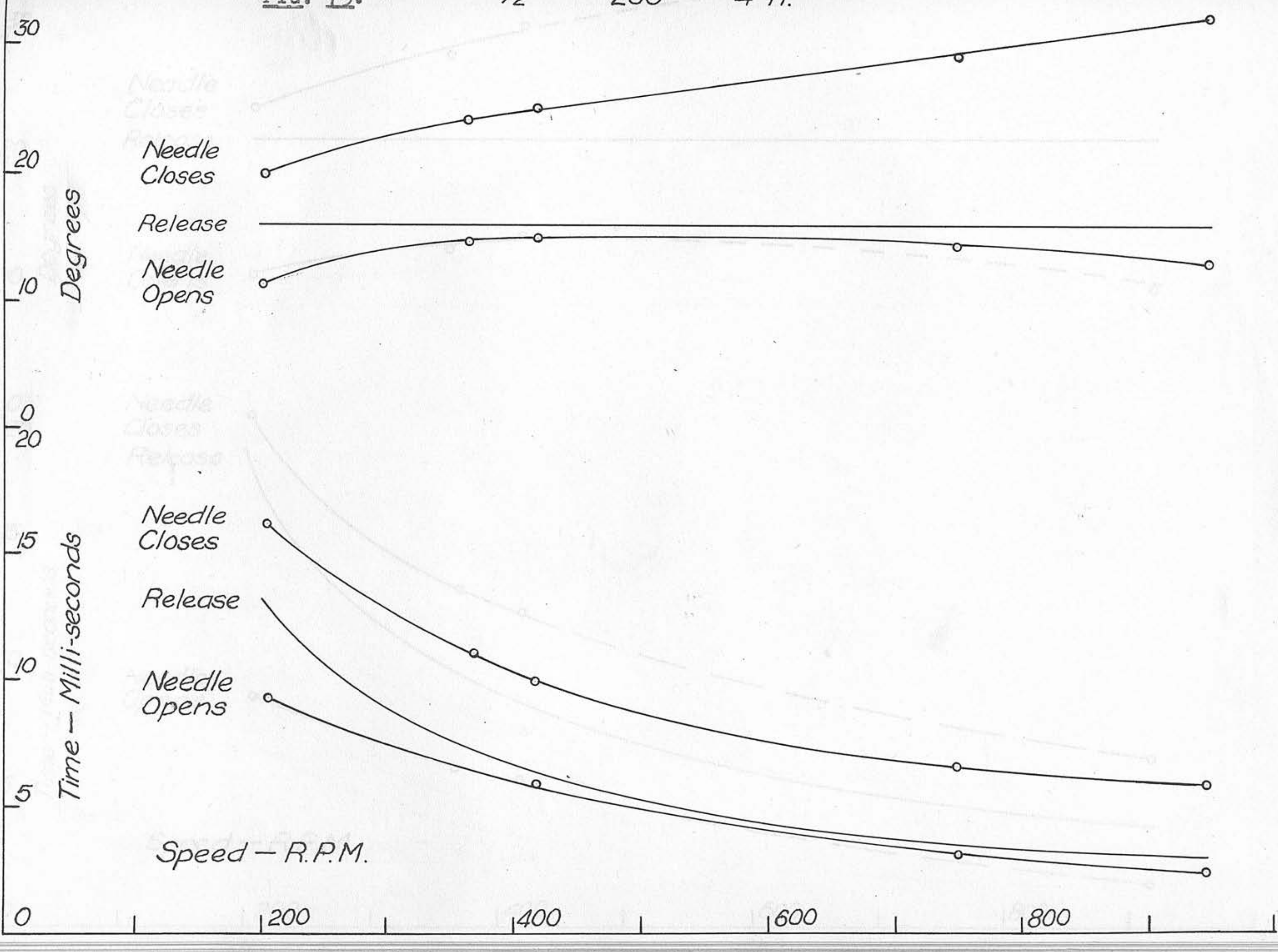


FIG. 76. 3/4 200 4 ft.

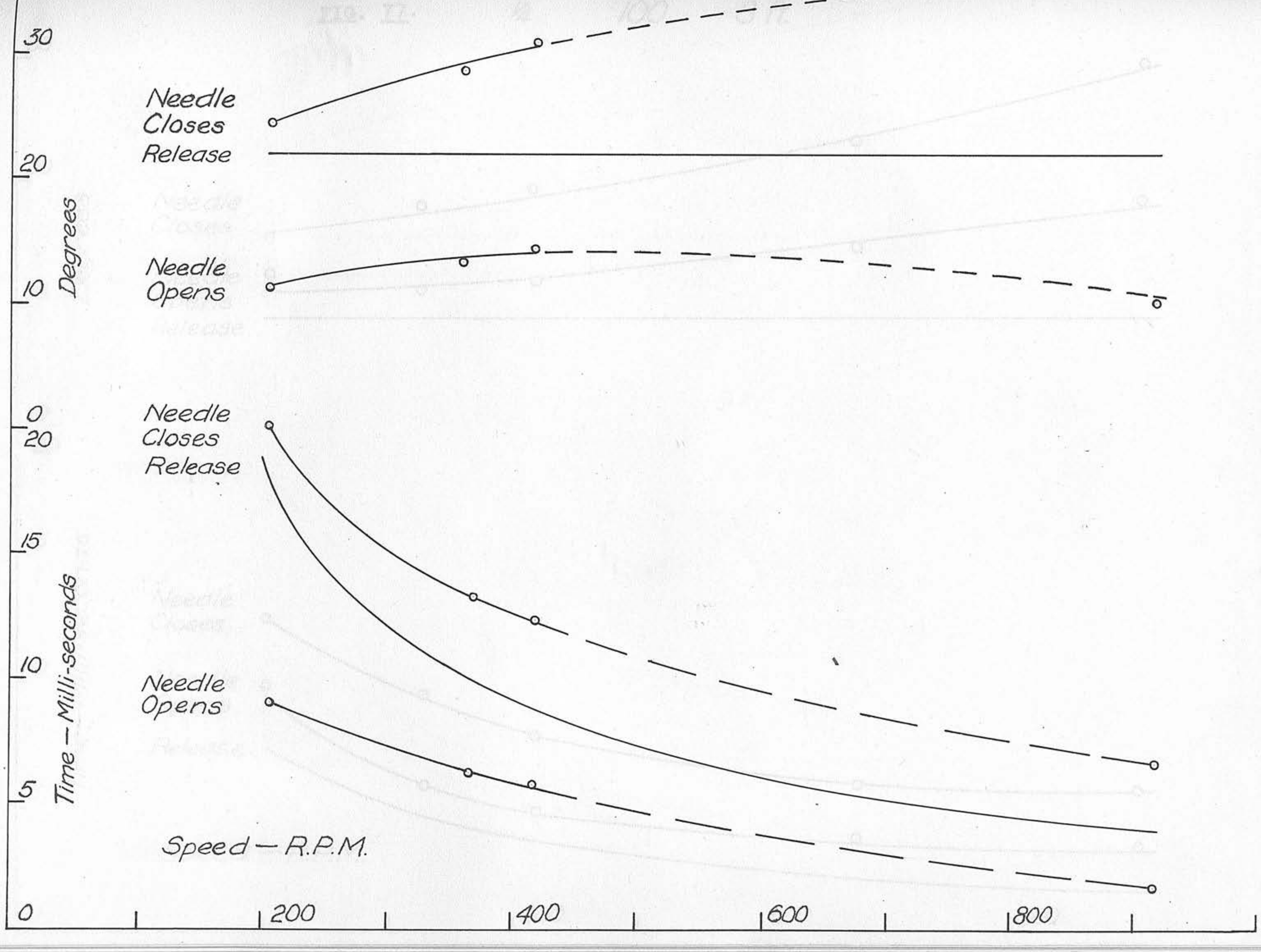


FIG. 77. ¼ 100 8 ft.

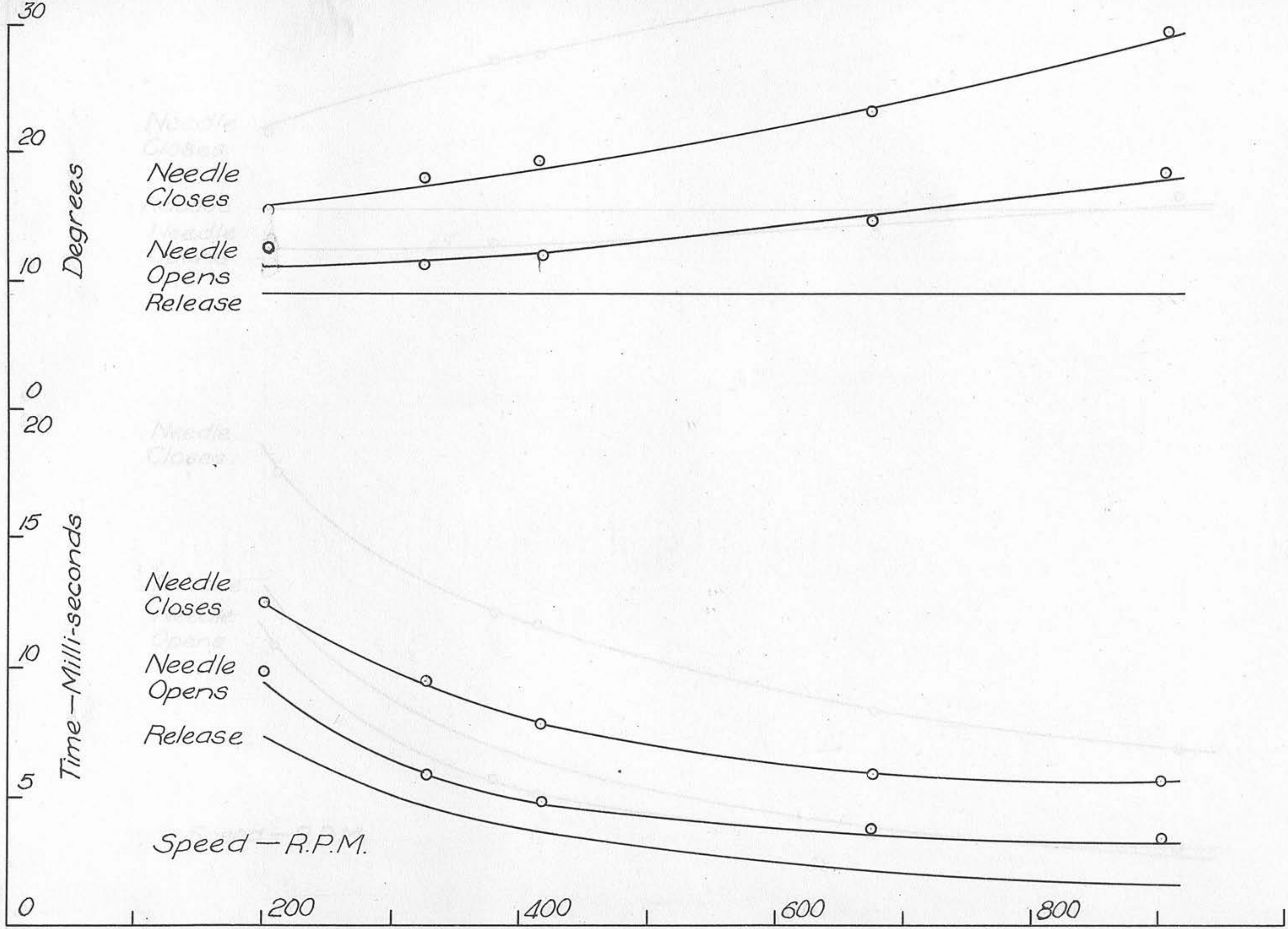


FIG. 78. ½ 100 8 ft.

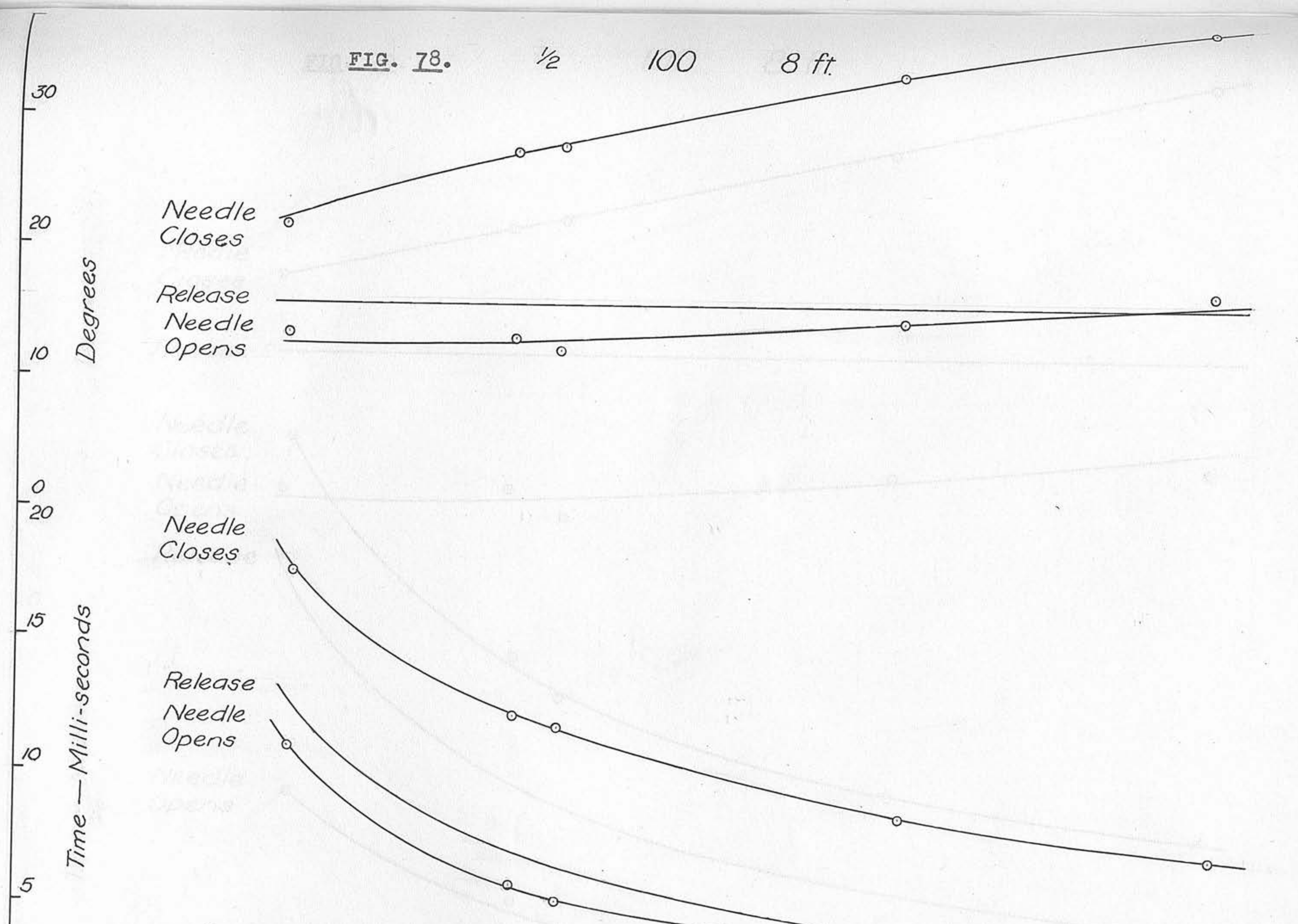


FIG. 79. 3/4 100 8 ft.

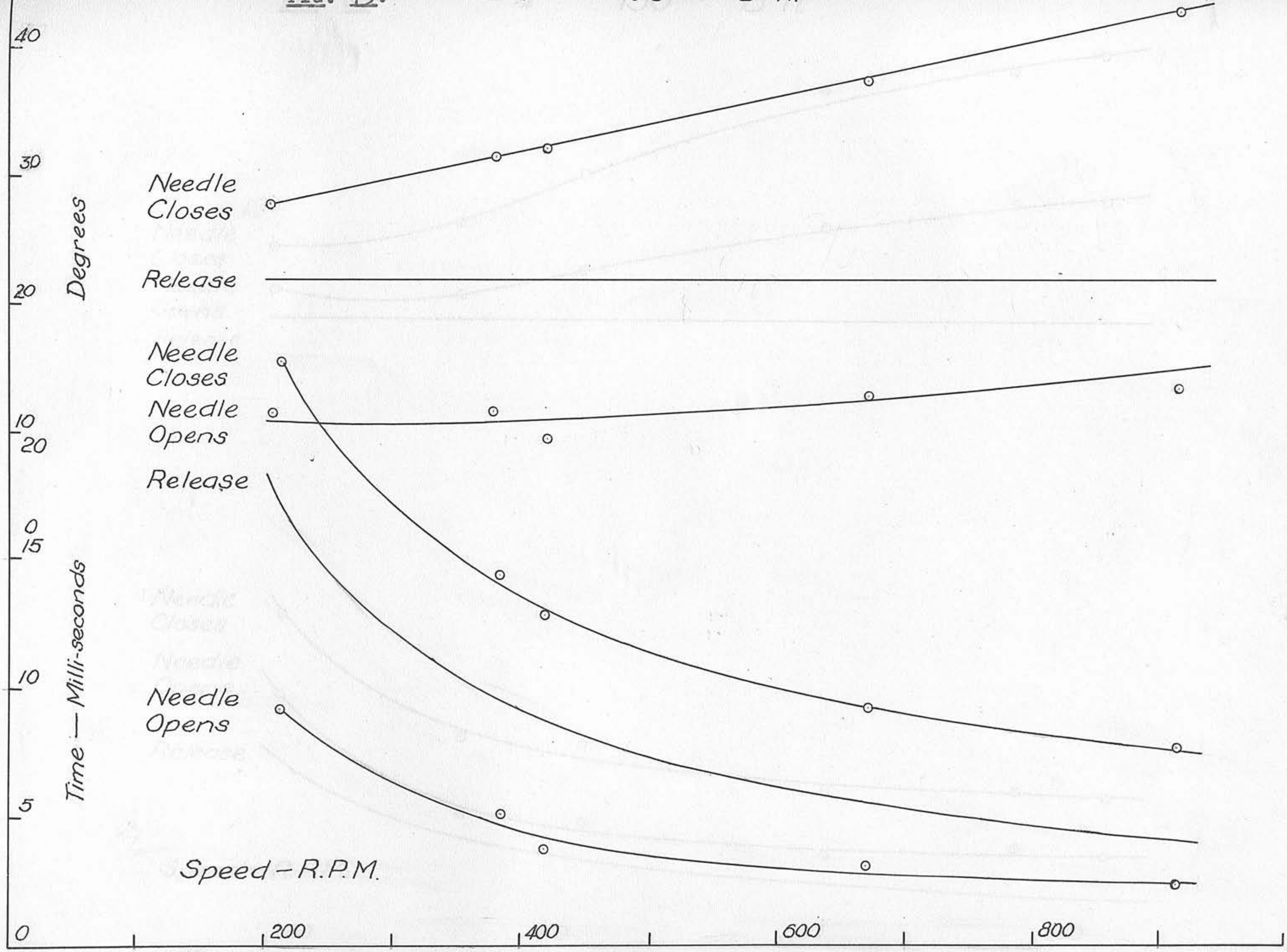


FIG. 80. ¼ 150 8 ft.

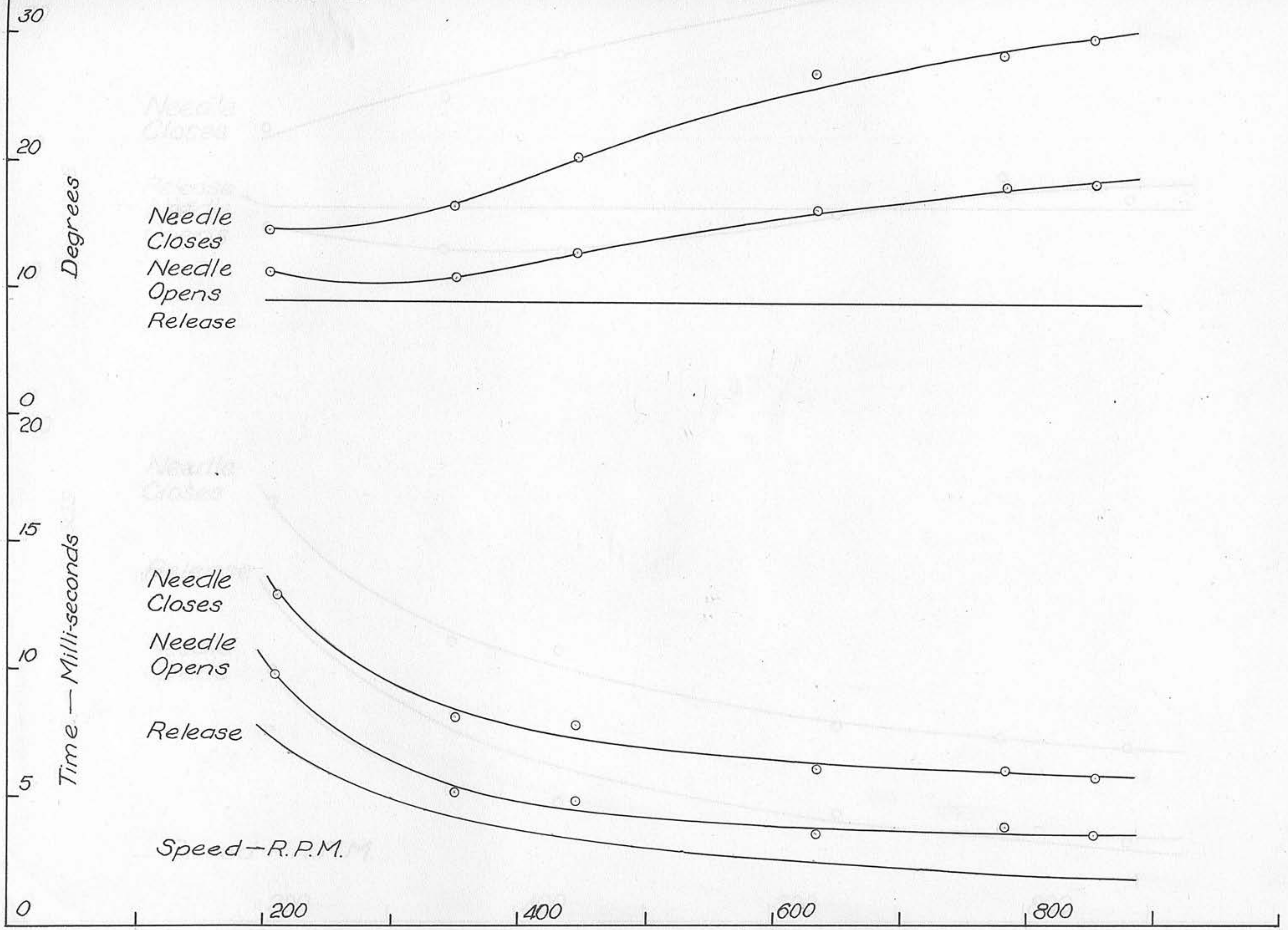


**FIG. 81.** ½ 150 8 ft.

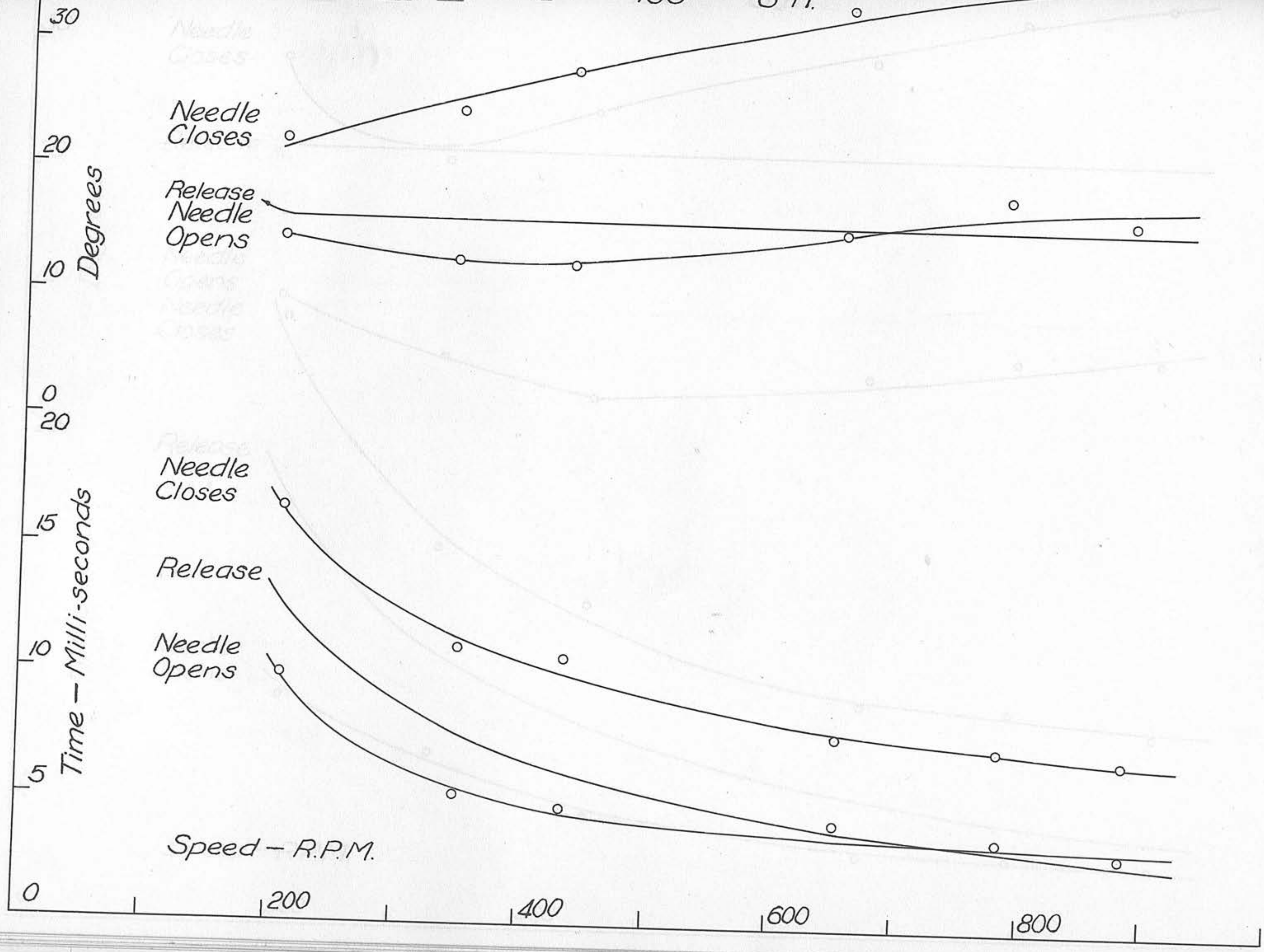


FIG. 82. 3/4 150 8 ft.

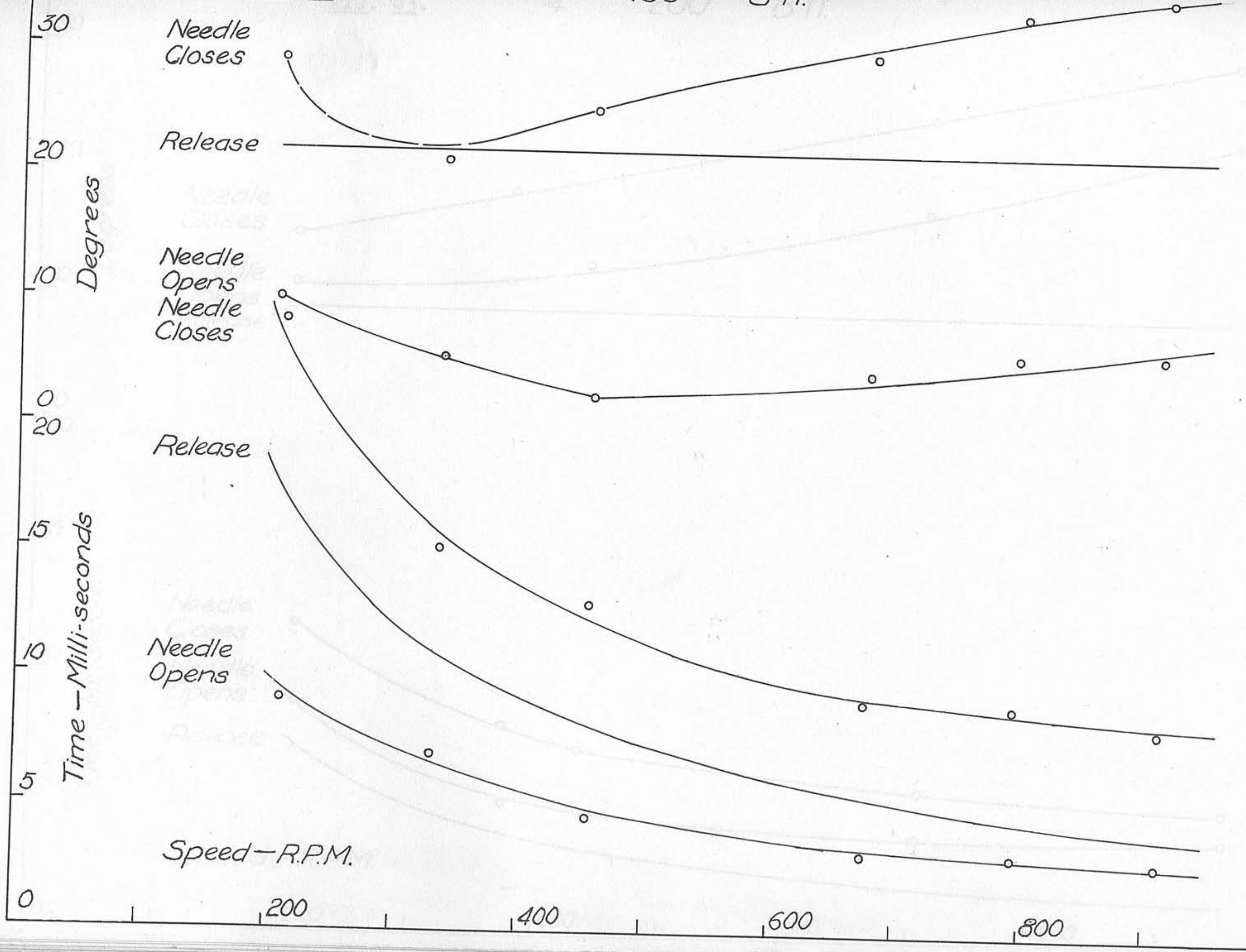


FIG. 83. 1/4 200 8 ft.

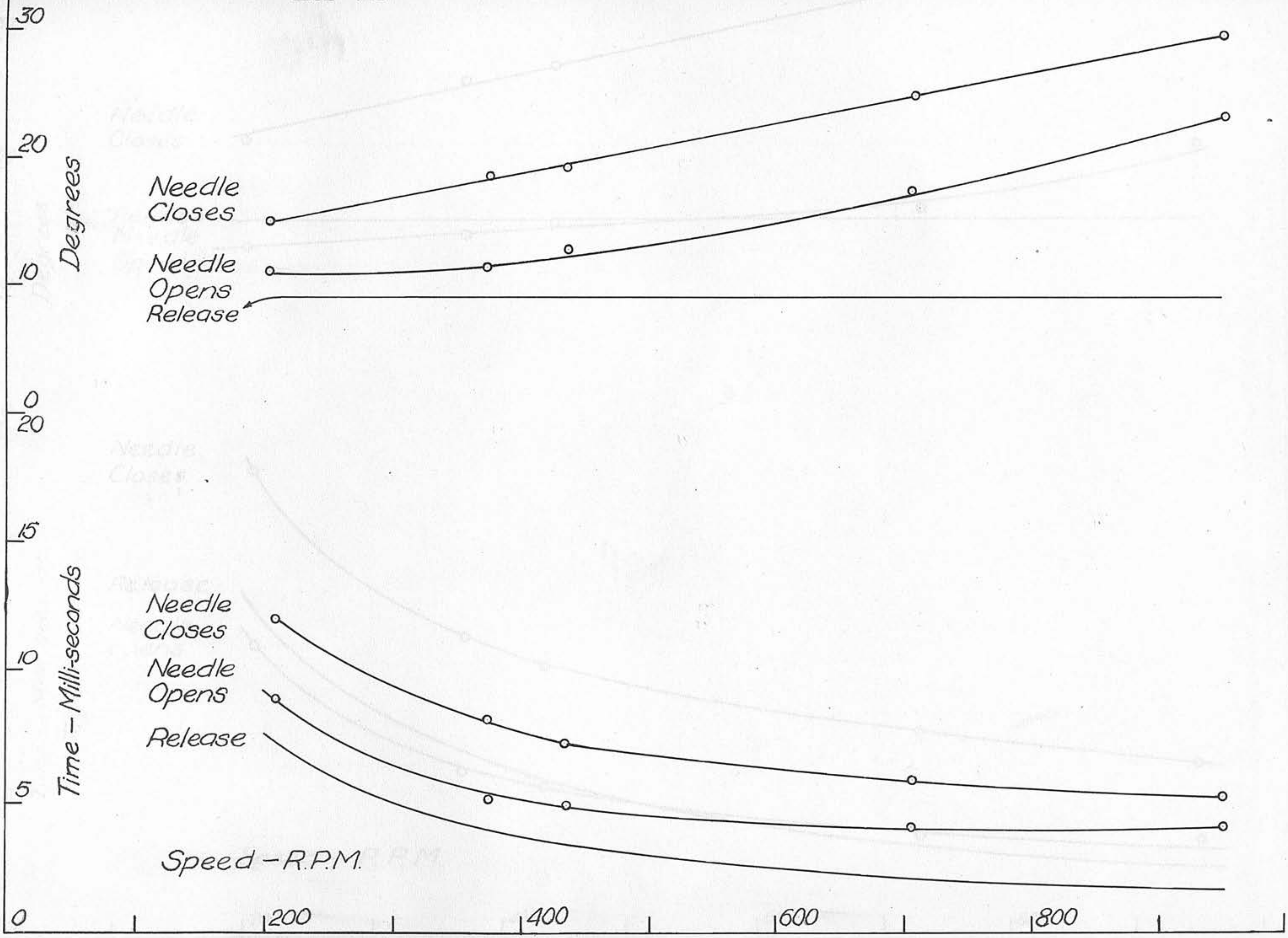


FIG. 84. ½ 200 8 ft.

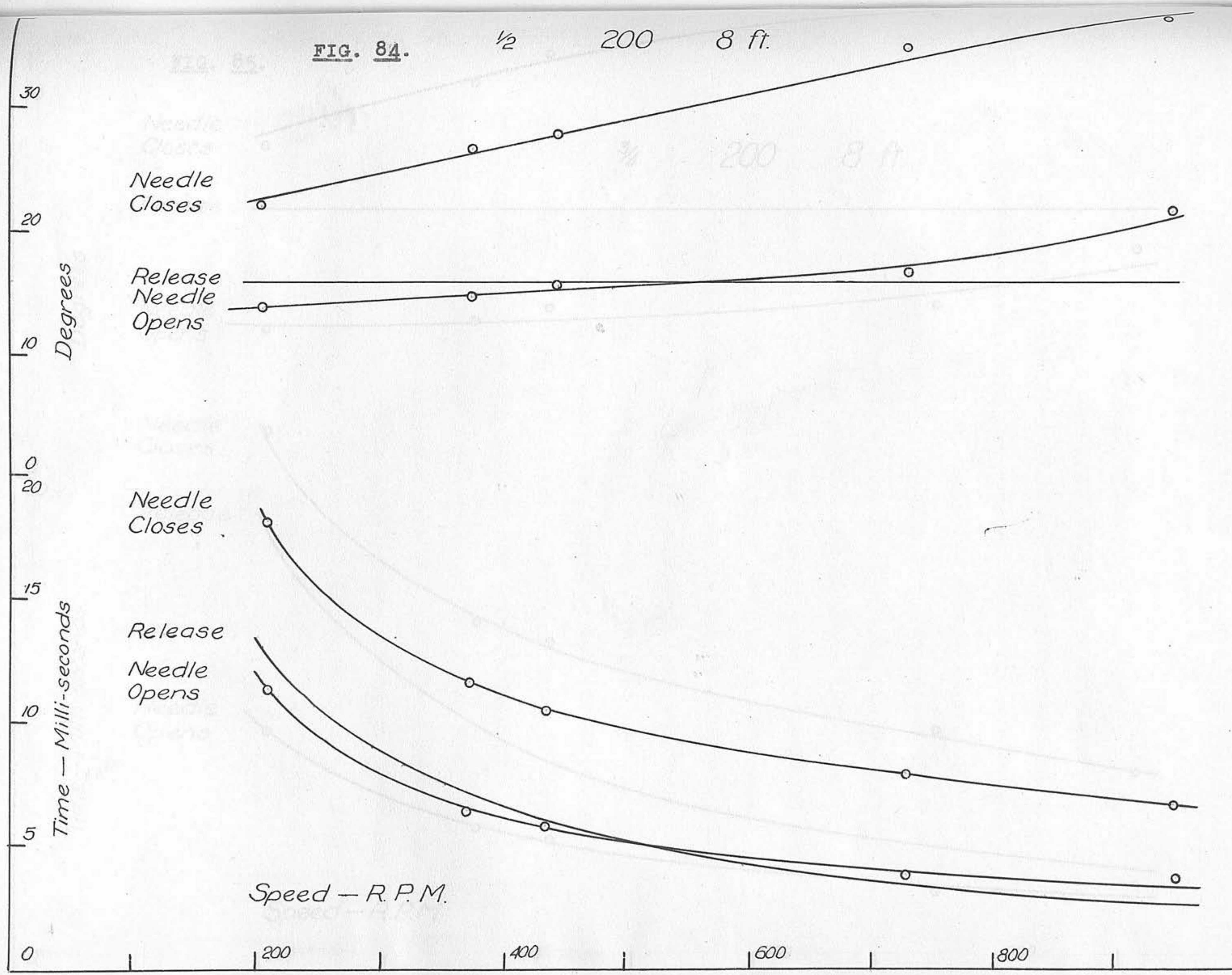
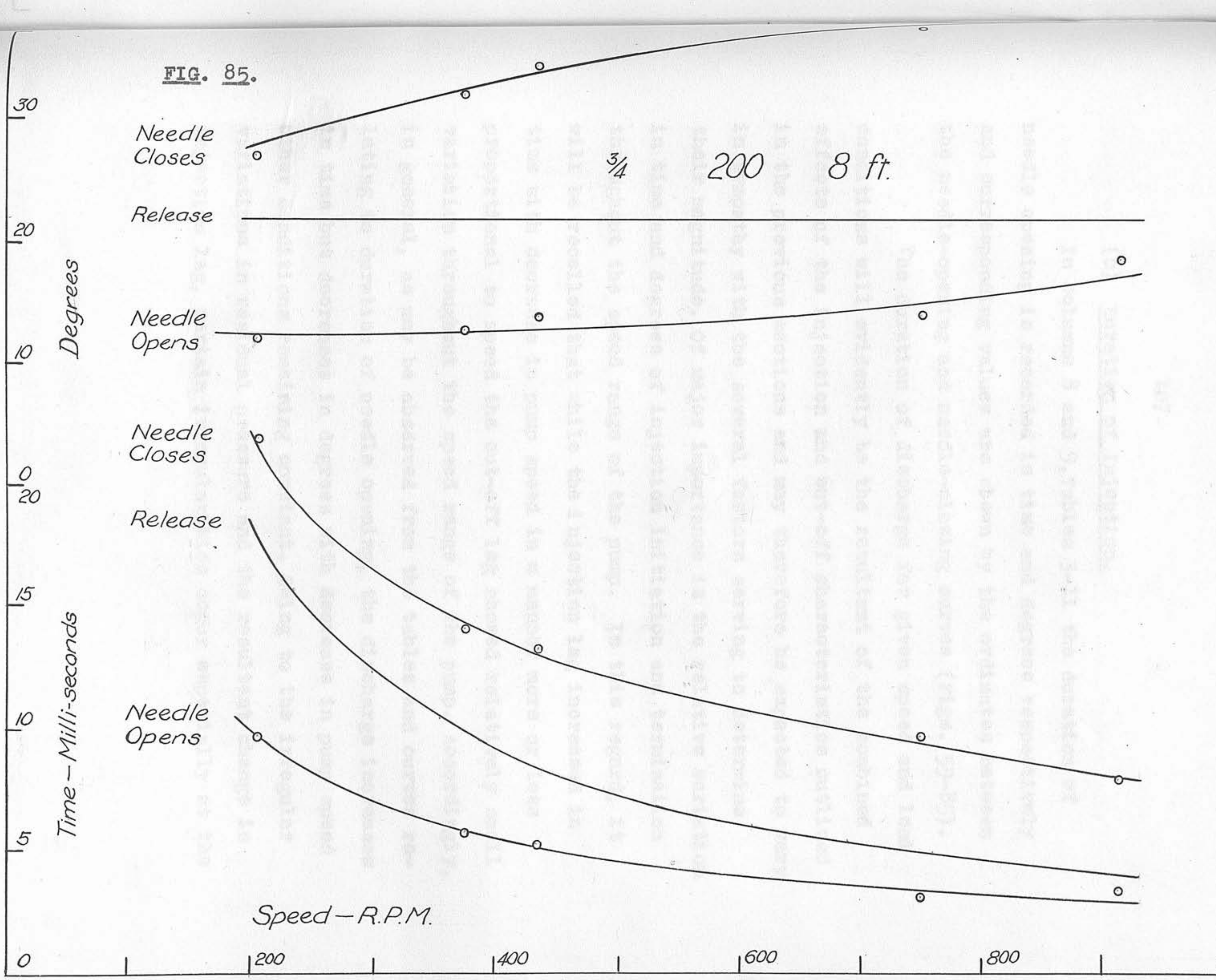


FIG. 85.



¾ 200 8 ft.
Needle Closes
Release
Needle Opens
Needle Closes
Release
Needle Opens
Degrees
Time – Milli-seconds
Speed – R.P.M.
30
20
10
0
20
15
10
5
0
200
400
600
800

(b) Duration of Injection.

In Columns 8 and 9, Tables 3-11 the duration of needle opening is recorded in time and degrees respectively and corresponding values are shown by the ordinates between the needle-opening and needle-closing curves (Figs. 59-85).

The duration of discharge for given speed and load conditions will evidently be the resultant of the combined effects of the injection and cut-off characteristics outlined in the previous sections and may therefore be expected to vary in sympathy with the several factors serving to determine their magnitude. Of major importance is the relative variation in time and degrees of injection initiation and termination throughout the speed range of the pump. In this regard, it will be recalled that while the injection lag increased in time with decrease in pump speed in a manner more or less proportional to speed the cut-off lag showed relatively small variation throughout the speed range of the pump. Accordingly, in general, as may be observed from the tables and curves relating to duration of needle opening, the discharge increases in time but decreases in degrees with decrease in pump speed other conditions remaining constant. Owing to the irregular variations in residual pressure and the resultant change in injection lag, certain irregularities occur especially at the lower speeds. These are however of minor consequence and the above general statement holds with reasonable fidelity throughout the entire speed range examined. The duration of discharge remains sensibly constant for increasing pipe length with minor fluctuations occurring under conditions of irregular opening and closing characteristics.

With increasing needle tension, the injection lag increases for a given pipe length while the cut-off lag remains comparatively unaffected. Accordingly, it is noted that in general the duration of discharge decreases with increasing needle tension for a given pipe length, the decrease beomcing less marked at the lower speeds for all pipe lengths examined.

(c) <u>Discharge</u>:

The discharge in cubic-millimetres per pump stroke under the variable conditions of operation is set out in Tables 12-14 and the corresponding values plotted against pump speed in Figs. 86 - 95.

Following upon the analysis of the duration of discharge in the previous section wherein it was observed that the duration of discharge was relatively independent of pipe-length and decreased with increasing needle tension, it might be assumed that the discharge would follow a similar sequence. There are, however, certain fundamental processes of energy distribution associated with the discharge which serve to modify this assumption. For a constant volume displacement at the pump, the resultant energy change in the fuel line will be divided between compressing and discharging the fuel and for a constant residual pressure, the mean pressure throughout the discharge period will decrease with increasing pipe length owing to the increased volume of the system. In a system which is fully relieved to atmospheric pressure between successive pump strokes, the discharge, against a given needle tension will therefore,neglecting for the moment the effect of pressure waves,decrease almost proportionally with increase in pipe length. With the system under investigation, the full relief to atmospheric pressure does not take place; instead, a residual pressure varying throughout the entire speed and load range exists in the system as the result of constant volume unloading under all speed and load conditions and in consequence of certain factors associated with needle movement and hydrostatic expansion of the fuel. Under these conditions, the energy input will obviously vary with the residual pressure being greater for greater values of the latter.Owing to the relatively large residual pressures obtaining with the longer pipe lengths, the deficiency in mean pressure on account of increased volume is greatly offset and although a slight decrease in discharge may in general be

TABLE 12.

2 FEET PIPE

DISCHARGE - $MM^3$ PER STROKE.

| THROTTLE | 100KG per $CM^2$ | | 150KG per $CM^2$ | | 200KG per $CM^2$ | |
|---|---|---|---|---|---|---|
| | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE |
| 1/4 | 888 | 23.80 | 900 | 23.50 | 876 | 23.20 |
| 1/2 | 882 | 81.80 | 906 | 81.00 | 894 | 79.50 |
| 3/4 | 900 | 136.80 | 906 | 136.50 | 858 | 136.10 |
| 1/4 | 678 | 22.58 | 708 | 22.30 | 678 | 20.20 |
| 1/2 | 666 | 81.60 | 690 | 81.00 | 678 | 79.00 |
| 3/4 | 708 | 136.90 | 690 | 136.40 | 666 | 136.00 |
| 1/4 | 411 | 19.10 | 426 | 18.10 | 420 | 17.70 |
| 1/2 | 402 | 80.00 | 420 | 78.20 | 408 | 77.40 |
| 3/4 | 420 | 136.50 | 408 | 136.00 | 402 | 135.50 |
| 1/4 | 312 | 19.30 | 330 | 17.80 | 312 | 16.10 |
| 1/2 | 306 | 78.60 | 318 | 77.40 | 315 | 76.20 |
| 3/4 | 318 | 135.50 | 310 | 134.60 | 300 | 133.30 |
| 1/4 | 192 | 19.30 | 216 | 16.20 | 204 | 14.60 |
| 1/2 | 186 | 76.00 | 198 | 74.70 | 192 | 73.00 |
| 3/4 | 190 | 132.50 | 192 | 131.60 | 186 | 131.00 |

TABLE 13.

4 FEET PIPE.

DISCHARGE - $MM^3$ PER STROKE.

| THROTTLE | 100KG per $CM^2$ | | 150KG per $CM^2$ | | 200KG per $CM^2$ | |
|---|---|---|---|---|---|---|
| | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE |
| 1/4 | 870 | 28.50 | 870 | 25.00 | 894 | 24.80 |
| 1/2 | 910 | 83.00 | 888 | 80.00 | 885 | 77.60 |
| 3/4 | 882 | 137.70 | 918 | 135.70 | 888 | 135.00 |
| 1/4 | 680 | 27.20 | 684 | 23.50 | 702 | 22.10 |
| 1/2 | 684 | 82.80 | 684 | 79.20 | 695 | 77.10 |
| 3/4 | 684 | 137.40 | 696 | 135.30 | 723 | 134.60 |
| 1/4 | 414 | 20.00 | 414 | 18.30 | 402 | 17.40 |
| 1/2 | 414 | 80.00 | 414 | 76.80 | 414 | 74.00 |
| 3/4 | 414 | 135.70 | 420 | 133.90 | 444 | 133.10 |
| 1/4 | 312 | 19.30 | 324 | 17.80 | 324 | 16.70 |
| 1/2 | 318 | 78.50 | 324 | 75.20 | 318 | 73.00 |
| 3/4 | 318 | 135.10 | 324 | 132.80 | 330 | 131.90 |
| 1/4 | 195 | 17.40 | 198 | 15.40 | 204 | 13.60 |
| 1/2 | 192 | 74.60 | 205 | 72.30 | 204 | 71.80 |
| 3/4 | 192 | 134.00 | 198 | 131.40 | 204 | 130.20 |

# TABLE 14.

8 FEET PIPE.

DISCHARGE - $MM^3$ PER STROKE.

| THROTTLE | 100KG per $CM^2$ | | 150KG per $CM^2$ | | 200KG per $CM^2$ | |
|---|---|---|---|---|---|---|
| | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE | SPEED R.P.M. | DISCHARGE |
| $\frac{1}{4}$ | 894 | 26.48 | 906 | 24.70 | 876 | 23.90 |
| $\frac{1}{2}$ | 900 | 82.00 | 906 | 76.60 | 876 | 75.70 |
| $\frac{3}{4}$ | 915 | 141.20 | 882 | 135.50 | 870 | 135.20 |
| $\frac{1}{4}$ | 690 | 25.93 | 702 | 24.40 | 684 | 22.50 |
| $\frac{1}{2}$ | 690 | 81.80 | 690 | 76.10 | 678 | 75.10 |
| $\frac{3}{4}$ | 670 | 141.20 | 702 | 135.50 | 672 | 134.80 |
| $\frac{1}{4}$ | 414 | 24.90 | 426 | 22.50 | 408 | 19.10 |
| $\frac{1}{2}$ | 414 | 78.38 | 408 | 74.30 | 408 | 72.80 |
| $\frac{3}{4}$ | 408 | 139.30 | 408 | 134.00 | 390 | 132.60 |
| $\frac{1}{4}$ | 312 | 20.50 | 330 | 19.60 | 318 | 16.30 |
| $\frac{1}{2}$ | 312 | 76.34 | 312 | 73.20 | 315 | 71.70 |
| $\frac{3}{4}$ | 306 | 139.30 | 315 | 133.00 | 306 | 132.10 |
| $\frac{1}{4}$ | 192 | 17,90 | 210 | 17.10 | 198 | 14.60 |
| $\frac{1}{2}$ | 198 | 76.42 | 198 | 72.00 | 198 | 70.70 |
| $\frac{3}{4}$ | 186 | 137.80 | 192 | 130.80 | 192 | 129.60 |

FIGS. 86 - 94.

DISCHARGE.

FIG. 86.

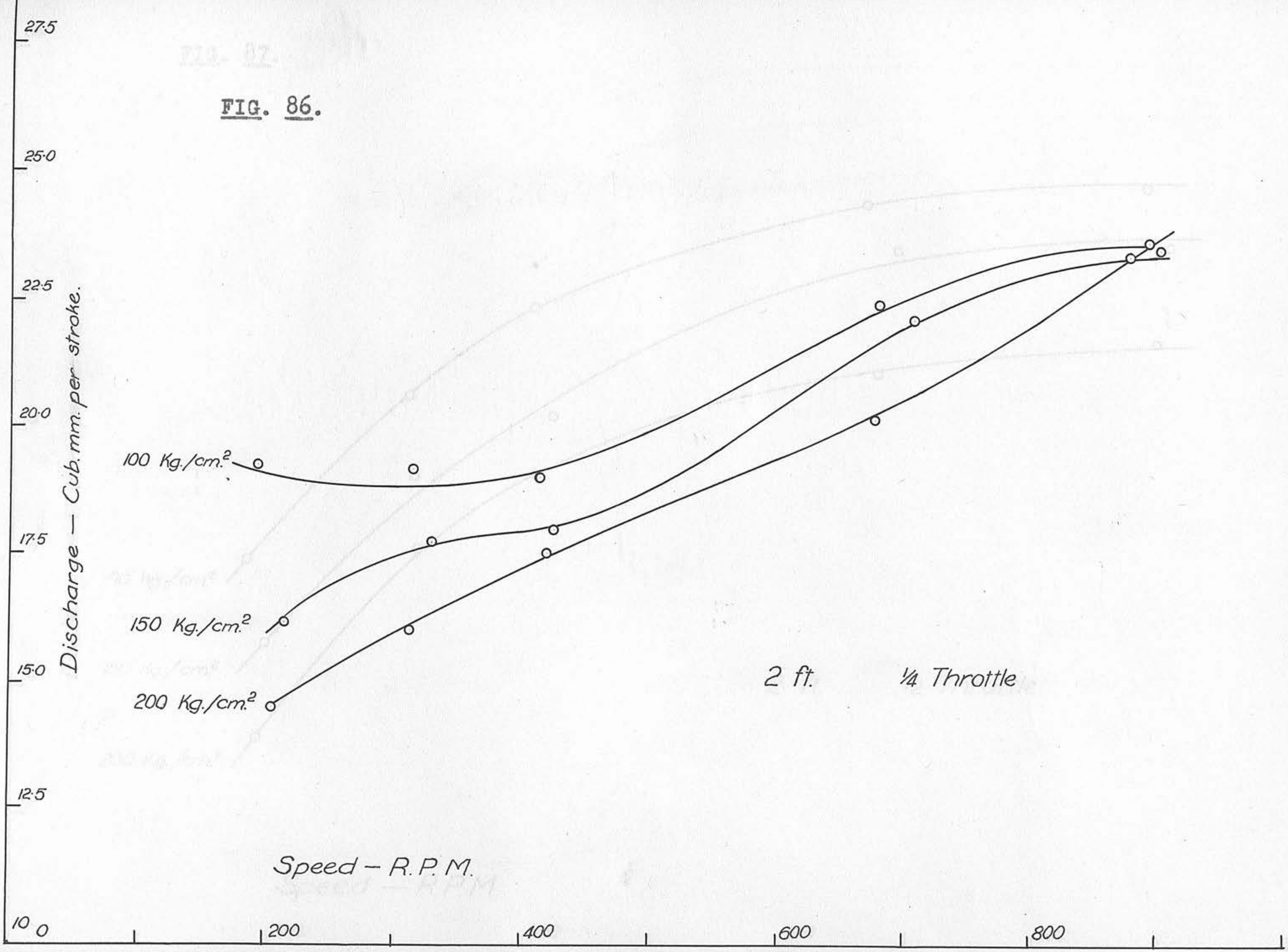
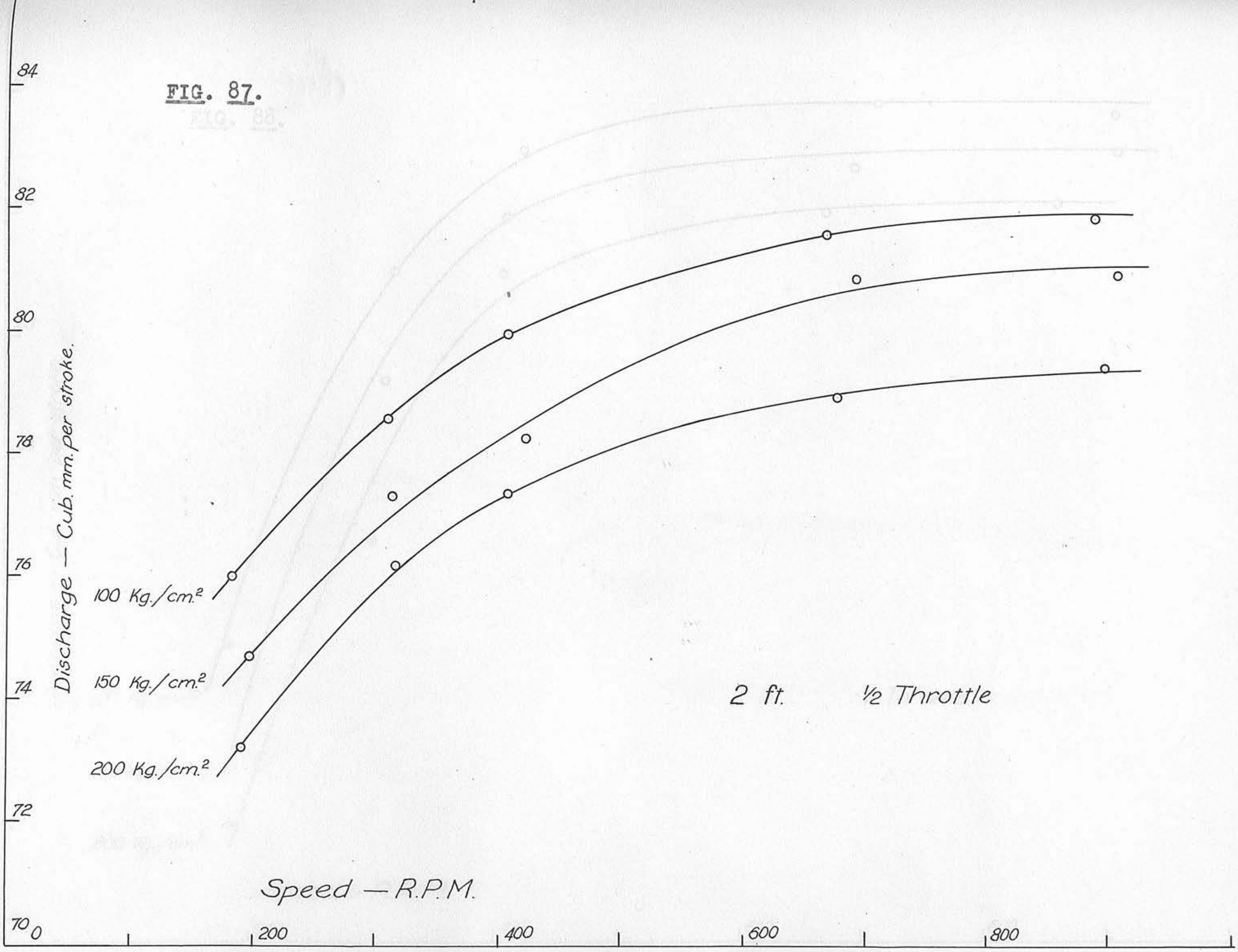
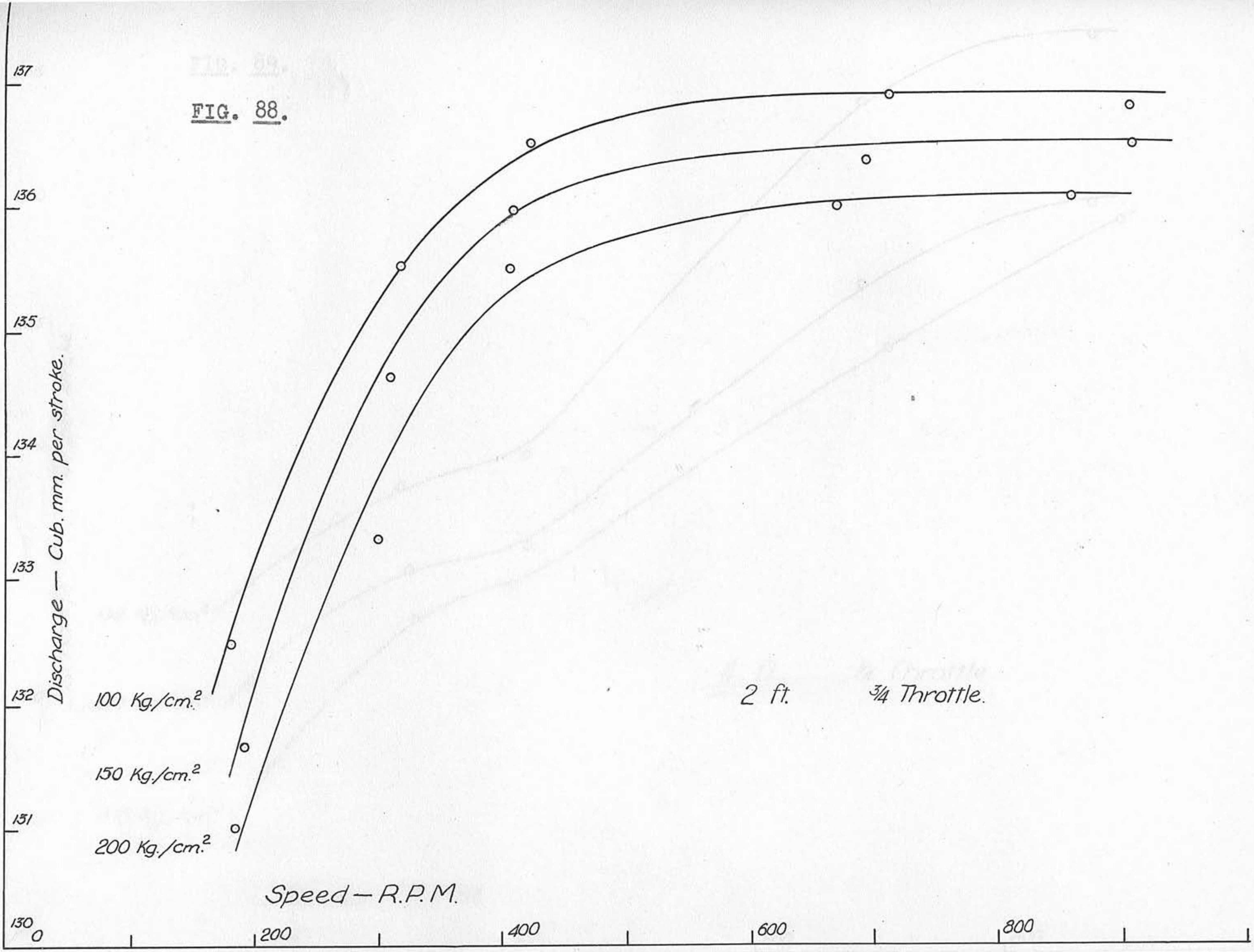

27·5
25·0
22·5
20·0
17·5
15·0
12·5
10 0
Discharge — Cub. mm. per stroke.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
2 ft.
¼ Throttle
Speed – R.P.M.
200
400
600
800

**FIG. 87.**



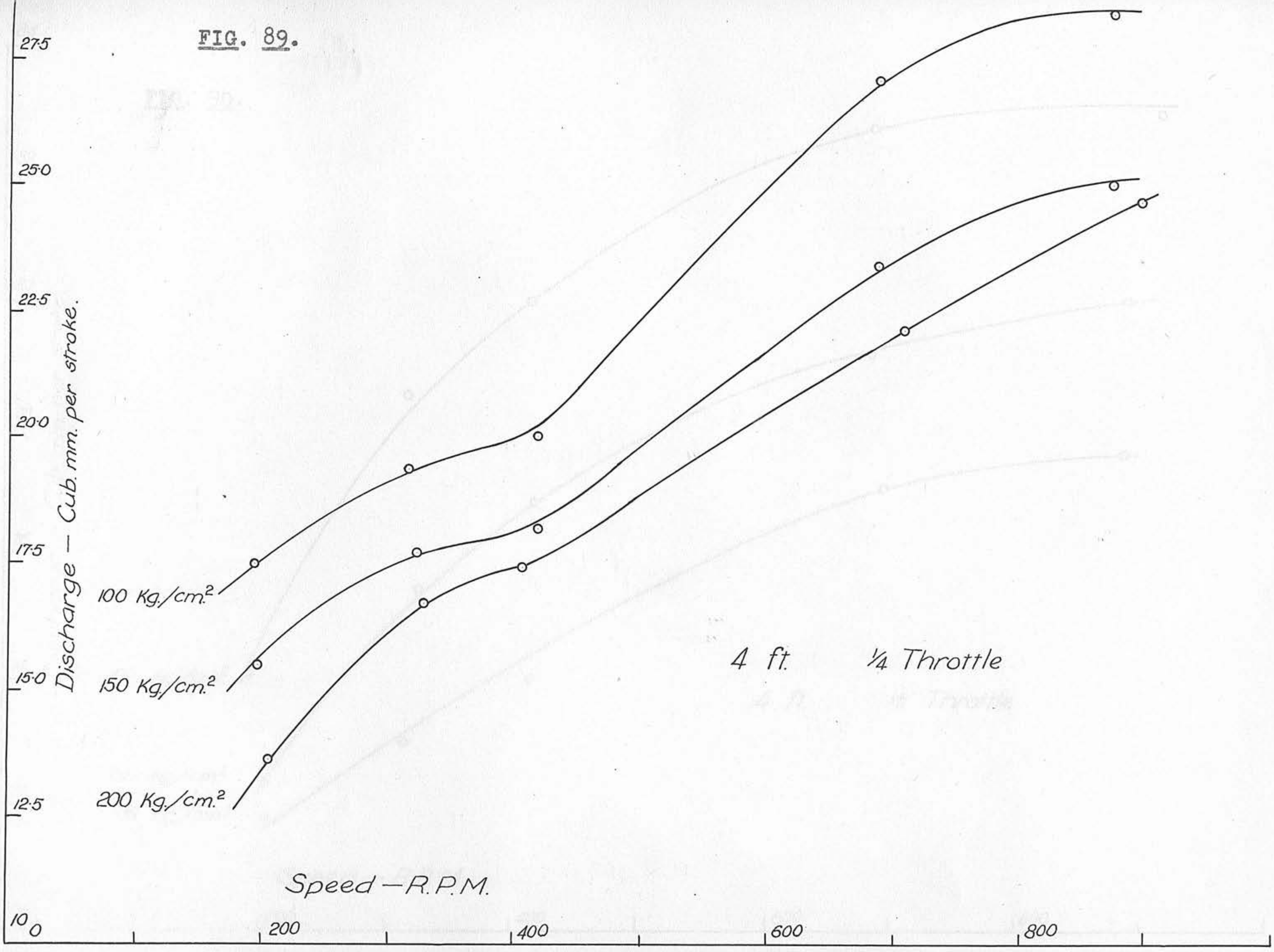
84
82
80
78
76
74
72
70
0
200
400
600
800
Discharge — Cub. mm. per stroke.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
2 ft.
½ Throttle
Speed — R.P.M.

FIG. 88.



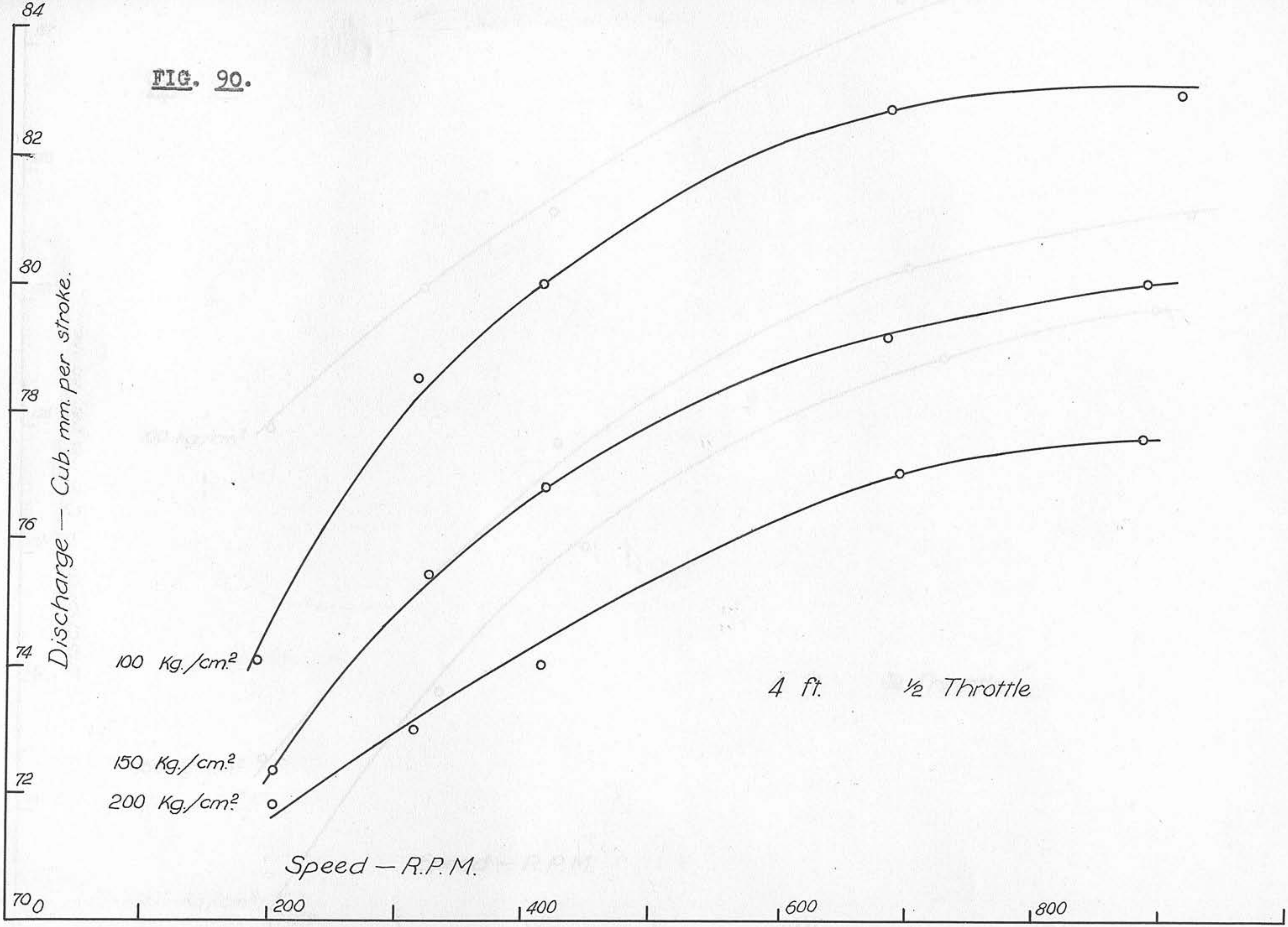
137
136
135
134
133
132
131
130
Discharge — Cub. mm. per stroke.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
2 ft.
3/4 Throttle.
Speed — R.P.M.
0
200
400
600
800

FIG. 89.



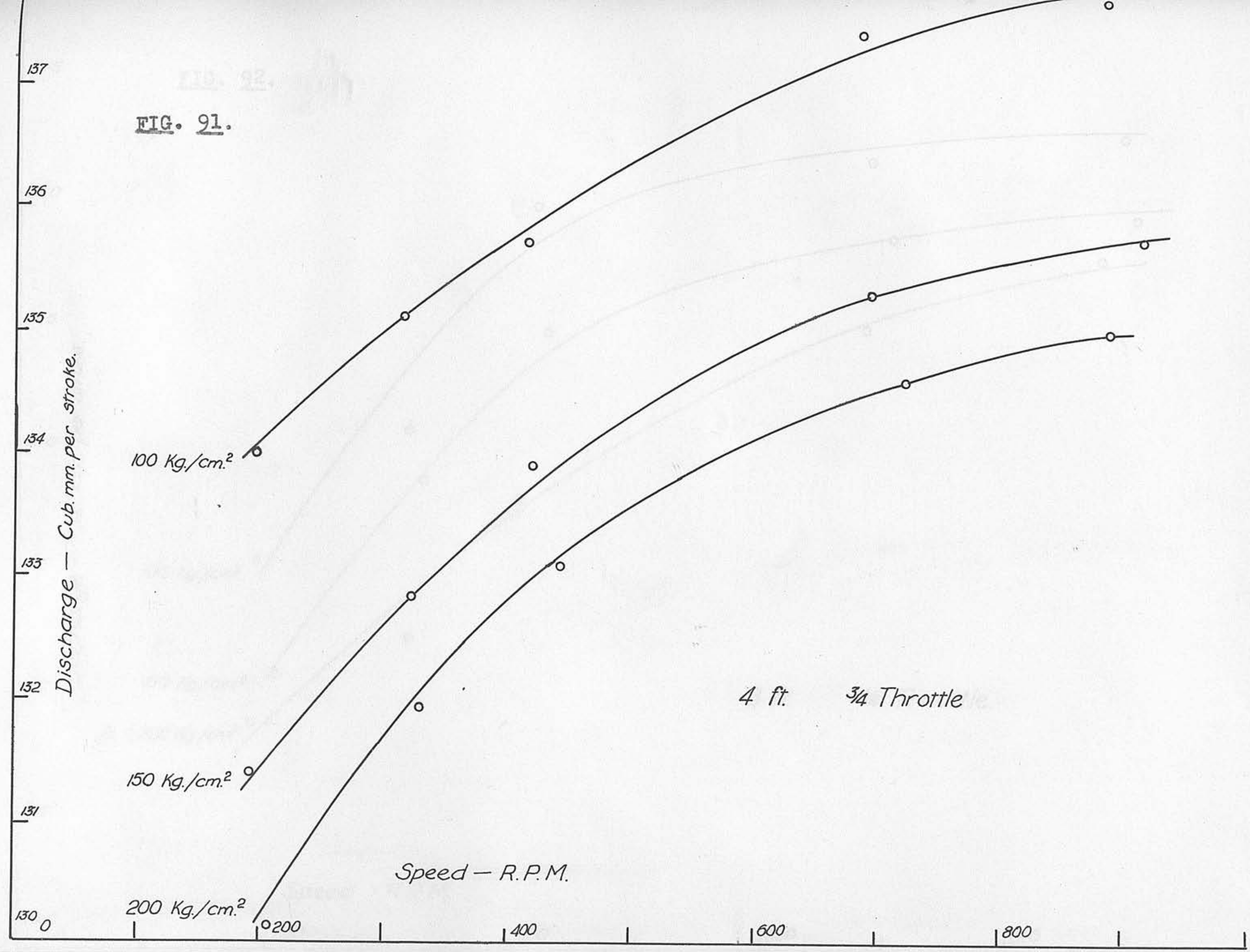
27·5
25·0
22·5
20·0
17·5
15·0
12·5
10
0
200
400
600
800
Discharge — Cub. mm. per stroke.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
4 ft.
¼ Throttle
Speed — R.P.M.

FIG. 90.



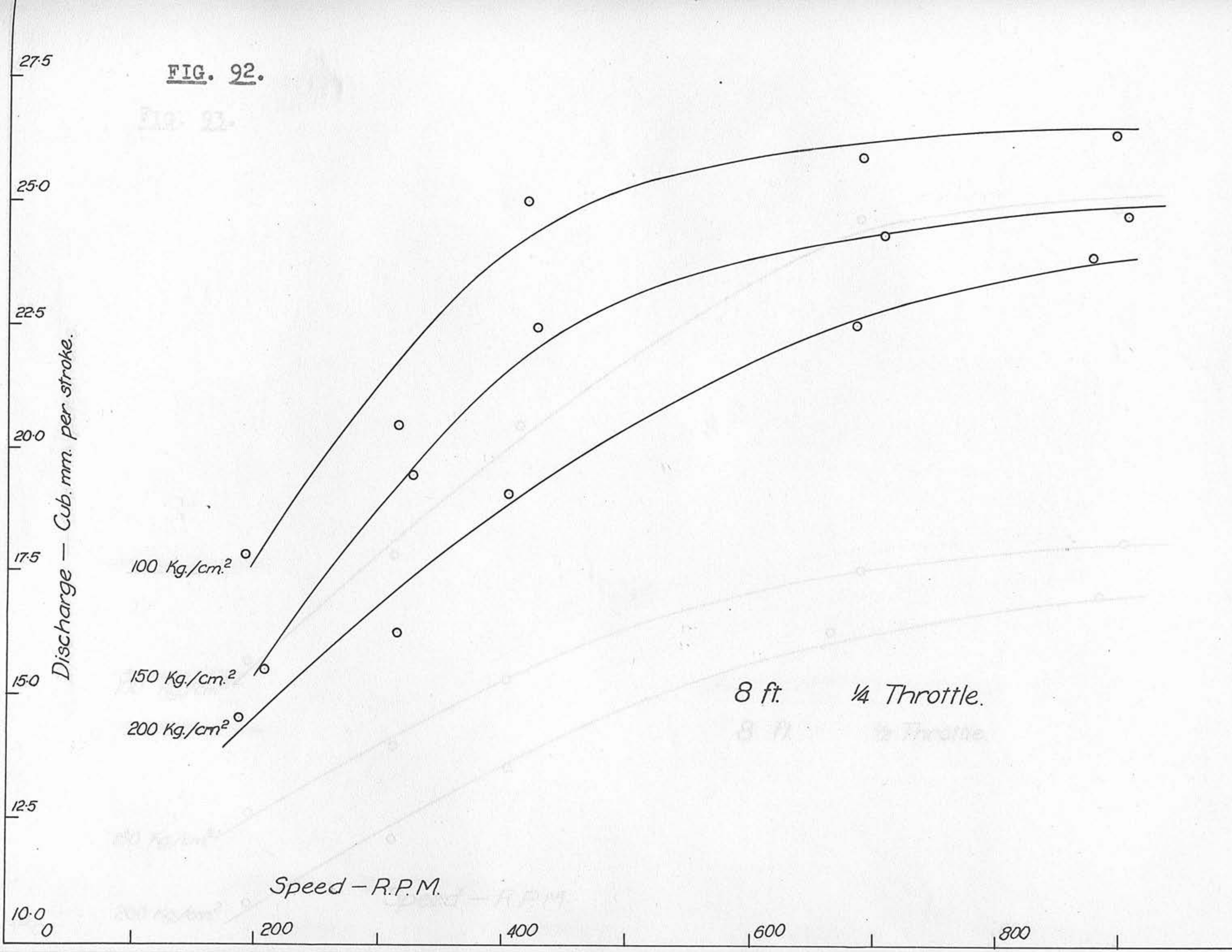
Discharge — Cub. mm. per stroke.
84
82
80
78
76
74
72
70
0
200
400
600
800
Speed — R.P.M.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
4 ft.
½ Throttle

FIG. 91.



Discharge — Cub. mm. per stroke.
137
136
135
134
133
132
131
130
0
200
400
600
800
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
4 ft.
¾ Throttle
Speed — R.P.M.

FIG. 92.



27·5
25·0
22·5
20·0
17·5
15·0
12·5
10·0
0
200
400
600
800
Discharge — Cub. mm. per stroke.
Speed – R.P.M.
100 Kg./cm.$^2$
150 Kg./cm.$^2$
200 Kg./cm$^2$
8 ft. ¼ Throttle.

FIG. 93.

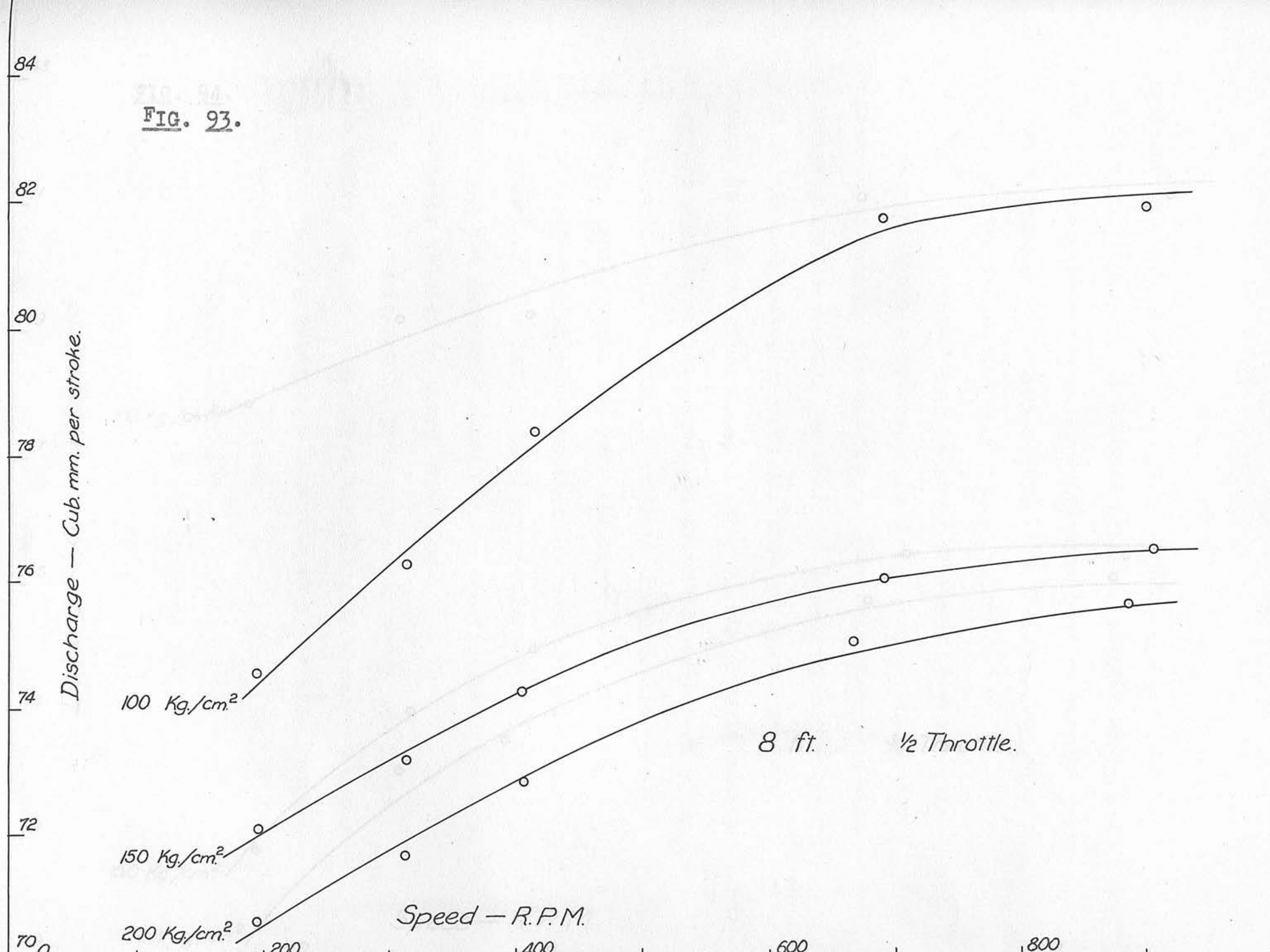
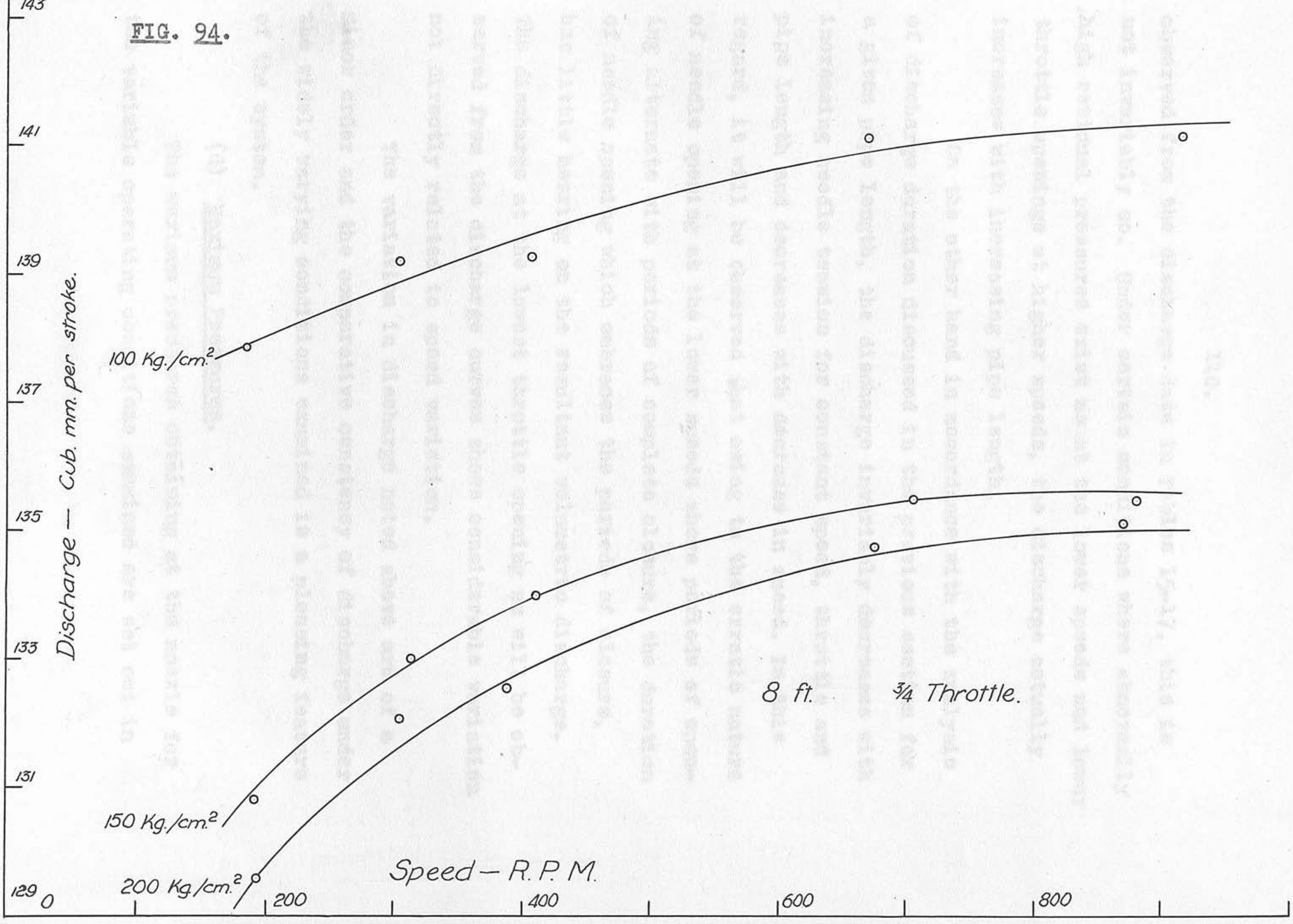

84
82
80
78
76
74
72
70
0
200
400
600
800
Discharge — Cub. mm. per stroke.
Speed — R.P.M.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
8 ft.
½ Throttle.

FIG. 94.



Discharge — Cub. mm. per stroke.
143
141
139
137
135
133
131
129
0
200
400
600
800
Speed — R. P. M.
100 Kg./cm.²
150 Kg./cm.²
200 Kg./cm.²
8 ft.
¾ Throttle.

observed from the discharge data in Tables 15-17, this is not invariably so. Under certain conditions where abnormally high residual pressures exist as at the lower speeds and lower throttle openings at higher speeds, the discharge actually increases with increasing pipe length.

On the other hand in accordance with the analysis of discharge duration discussed in the previous section for a given pipe length, the discharge invariably decreases with increasing needle tension for constant speed, throttle and pipe length and decreases with decrease in speed. In this regard, it will be observed that owing to the erratic nature of needle opening at the lower speeds where periods of opening alternate with periods of complete closure, the duration of needle opening which embraces the periods of closure, has little bearing on the resultant volumetric discharge. The discharge at the lowest throttle opening as will be observed from the discharge curves shows considerable variation not directly related to speed variation.

The variation in discharge noted above are of a minor order and the comparative constancy of discharge under the widely varying conditions examined is a pleasing feature of the system.

(d) Maximum Pressures.

The maximum pressures obtaining at the nozzle for the variable operating conditions examined are set out in

Col. 12, Tables 3-11 and the trend of pressure rise with speed shown in Figs. 96 - 104.

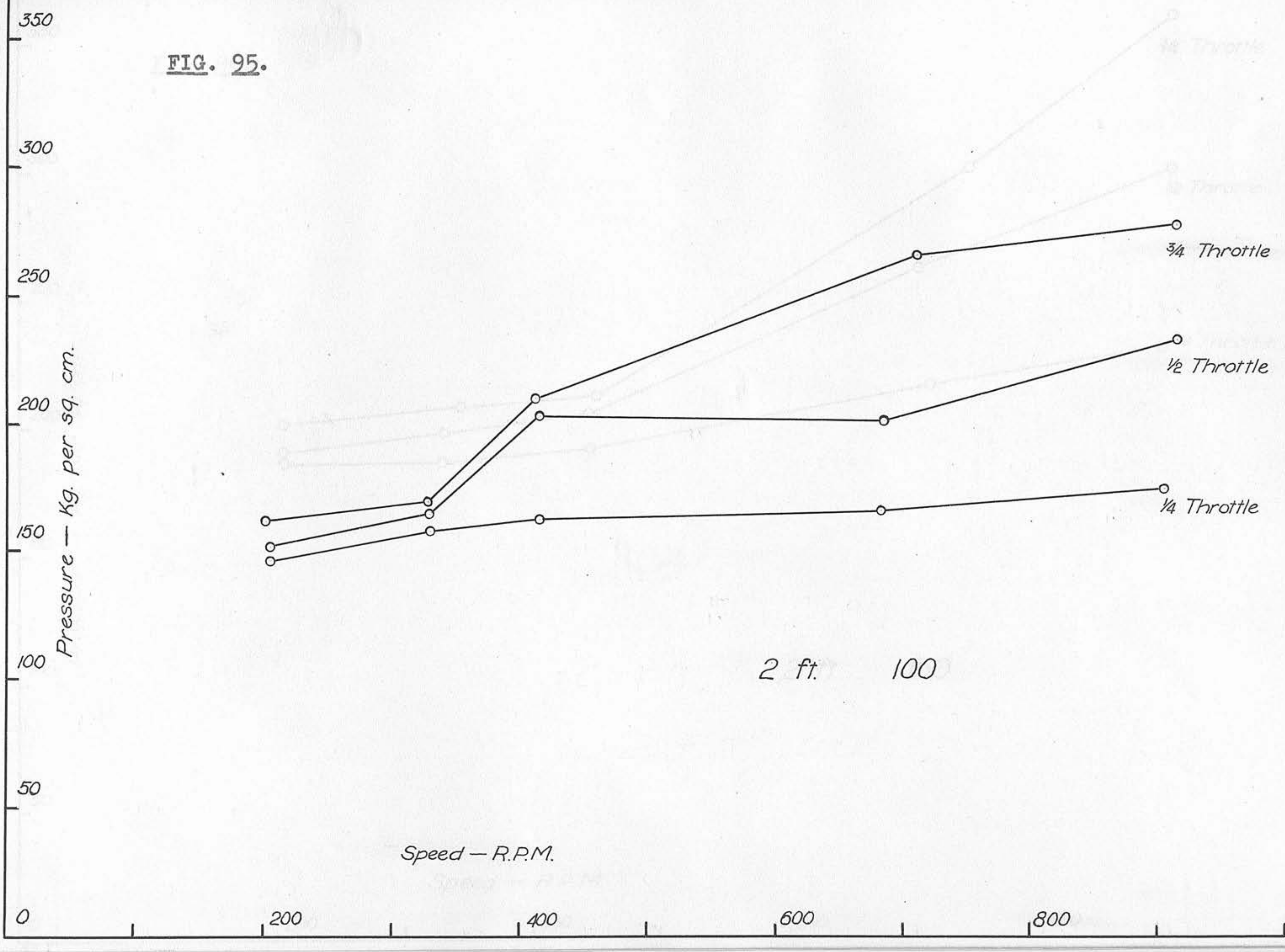
For all throttle openings at the lowest speed (200 r.p.m.) and for the ¼ throttle opening at all speeds, the maximum pressure at the nozzle does not rise above that obtaining behind the needle at the instant of opening. As may be observed from Figs. 96-104, this shows a slight but fairly uniform increase with increasing pump speed. At speeds in excess of 300 r.p.m. (nominal) and for throttle openings greater than ¼, the maximum pressures at the nozzle increase though not uniformly with increase in speed and throttle opening. The period during which higher pressures exist behind the needle also increases with speed and thus the quality of the injection on account of finer atomisation is improved.

The rate of pressure increase with speed increases with increasing throttle as shown by the steeper slope of the curves (Figs. 96-104) and although some variation is noticeable, the same holds true in general for pressure increase with increasing throttle at constant speed as shown in Figs. 104-106.

FIGS. 95 - 103.

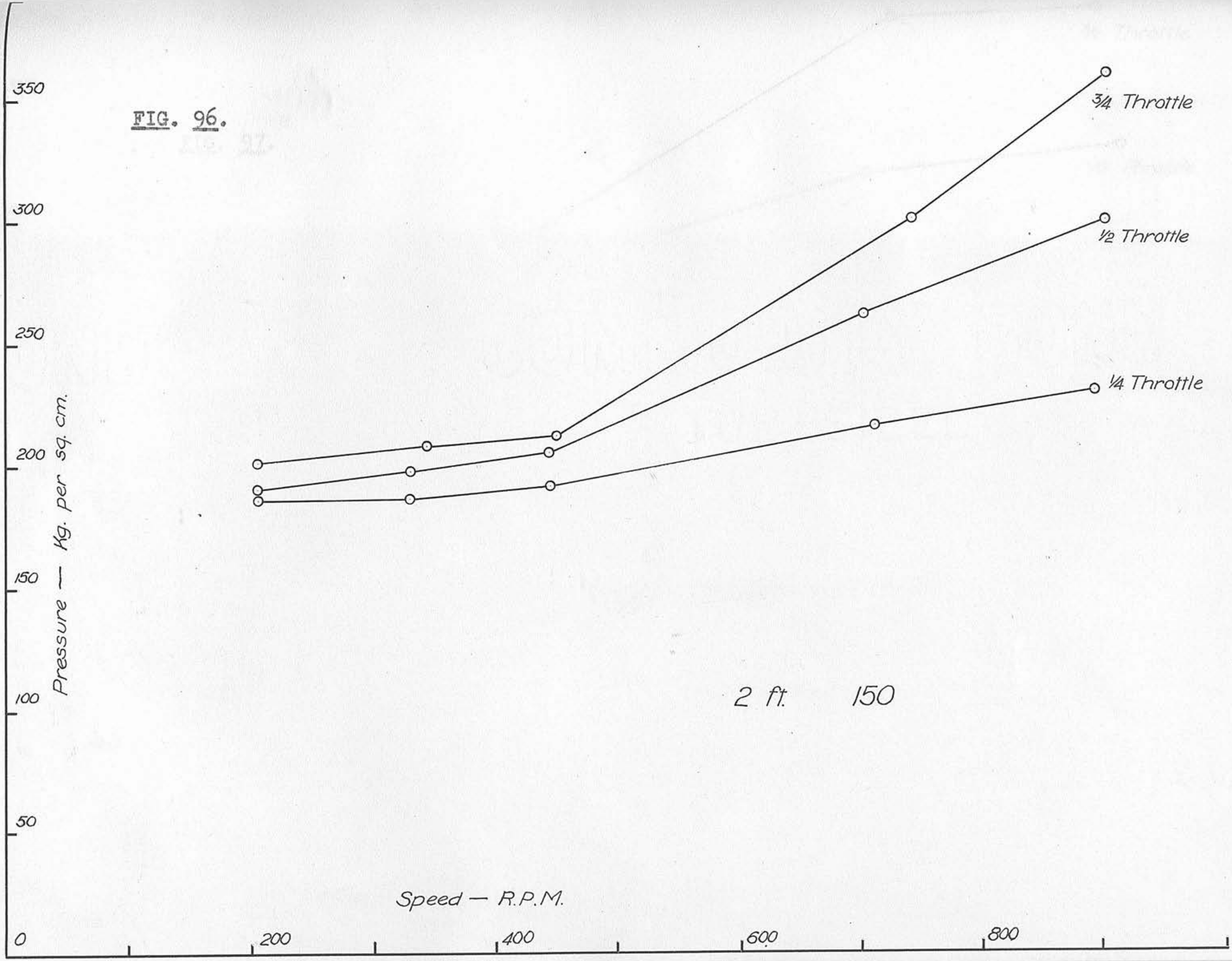
MAXIMUM PRESSURES.

FIG. 95.



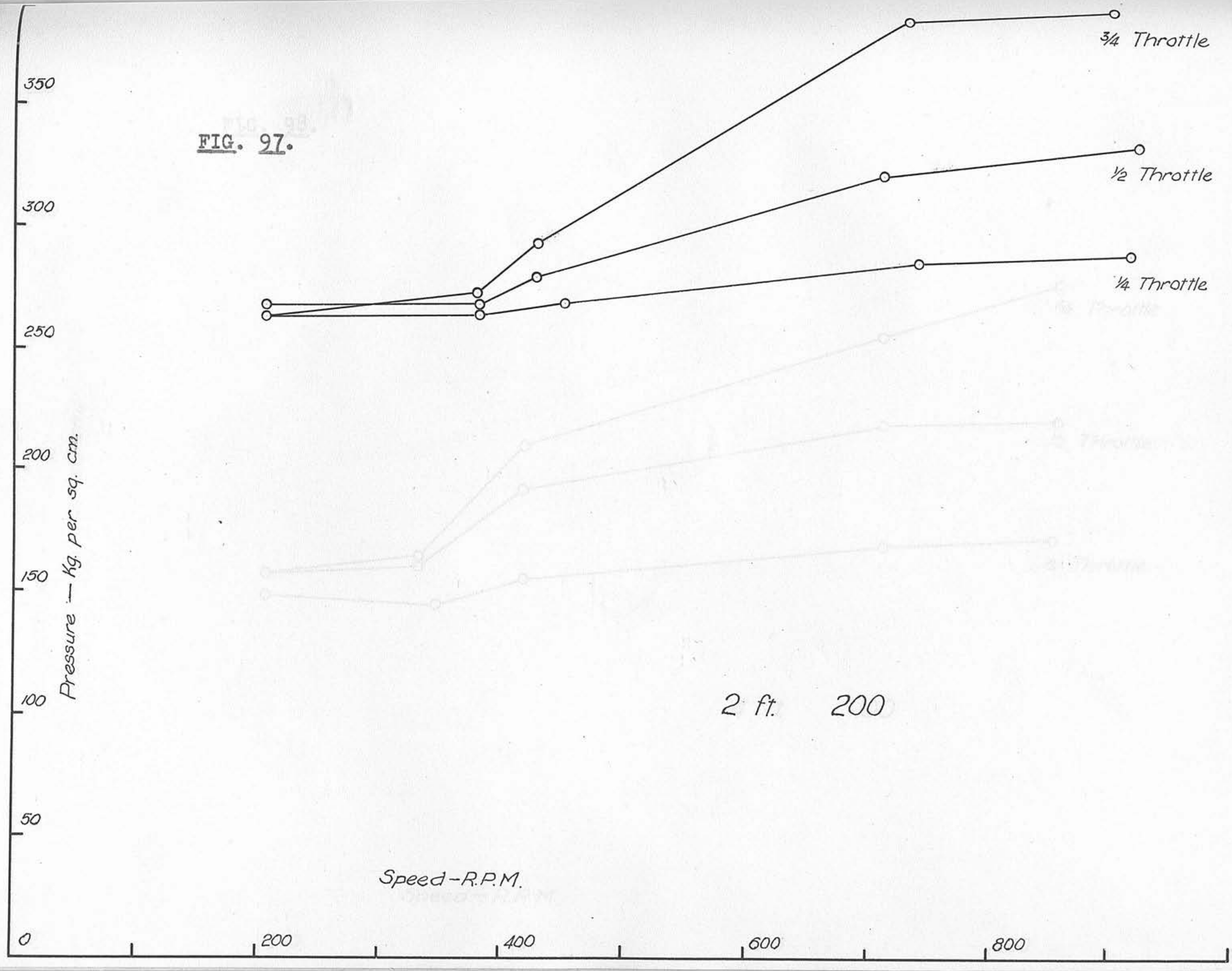
350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
¾ Throttle
½ Throttle
¼ Throttle
2 ft. 100
Speed — R.P.M.
200
400
600
800

FIG. 96.



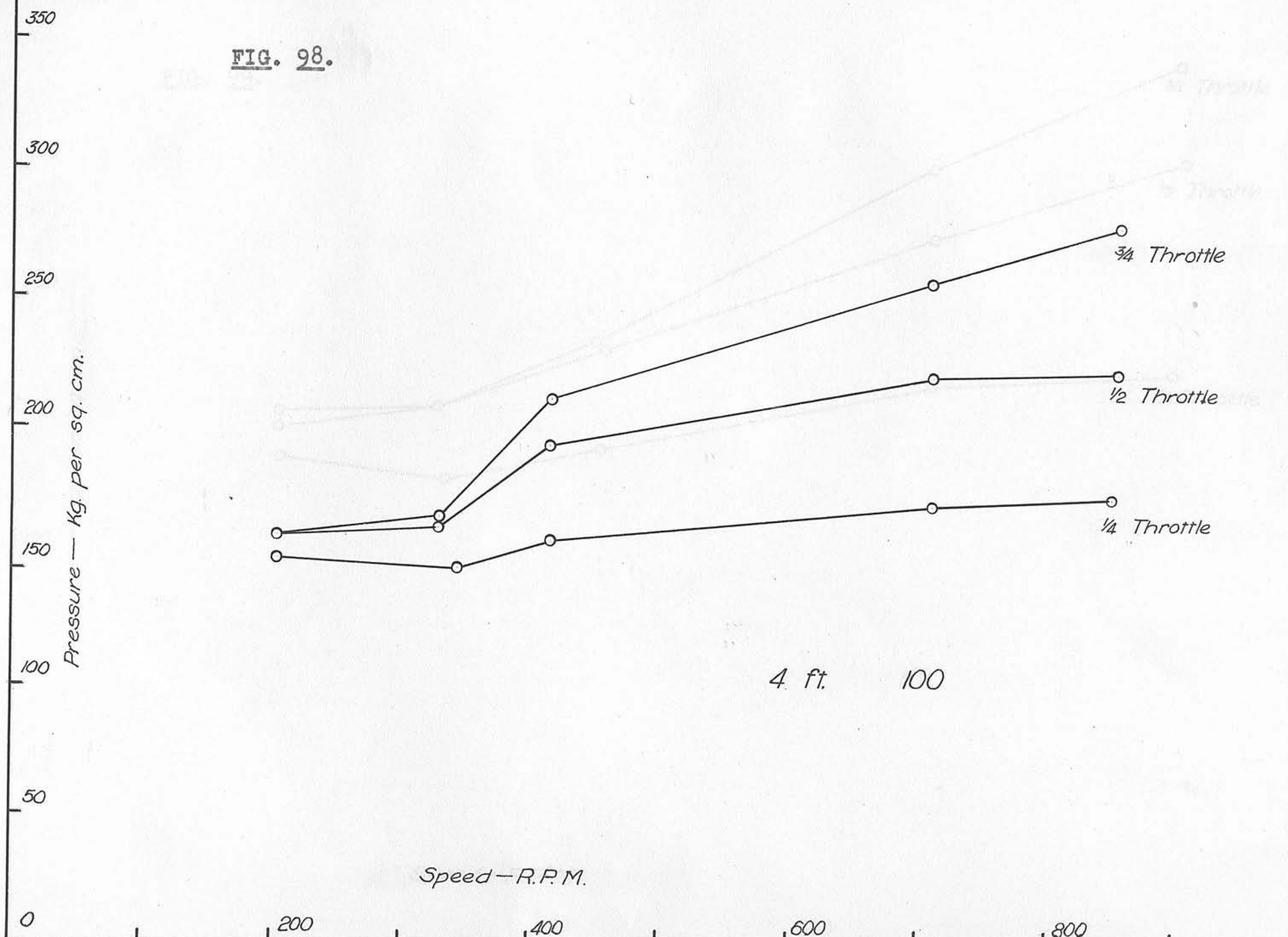
350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
200
400
600
800
Speed — R.P.M.
3/4 Throttle
1/2 Throttle
1/4 Throttle
2 ft.
150

FIG. 97.



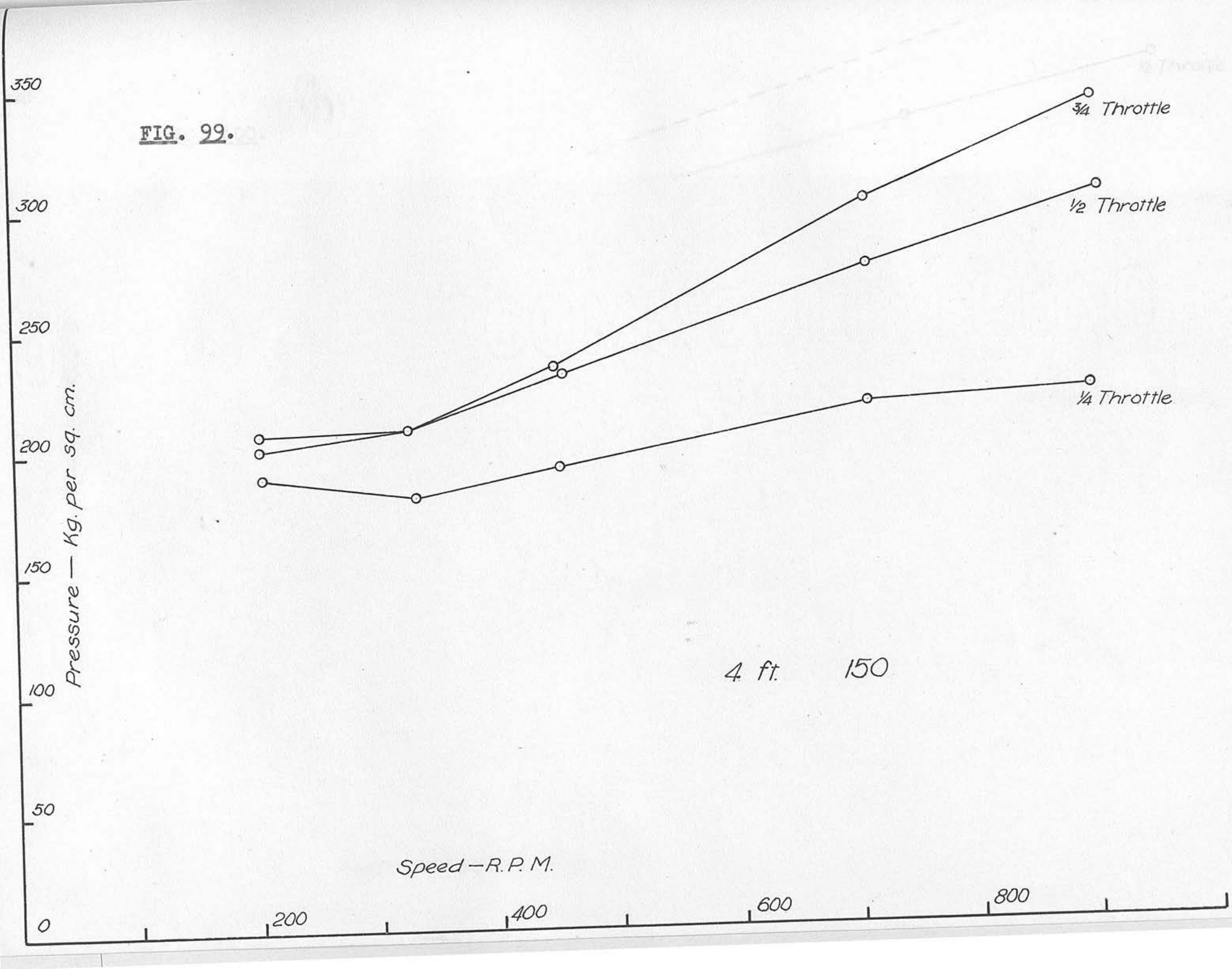
350
300
250
200
150
100
50
0
Pressure — kg. per sq. cm.
200
400
600
800
Speed-R.P.M.
3/4 Throttle
1/2 Throttle
1/4 Throttle
2 ft. 200

FIG. 98.



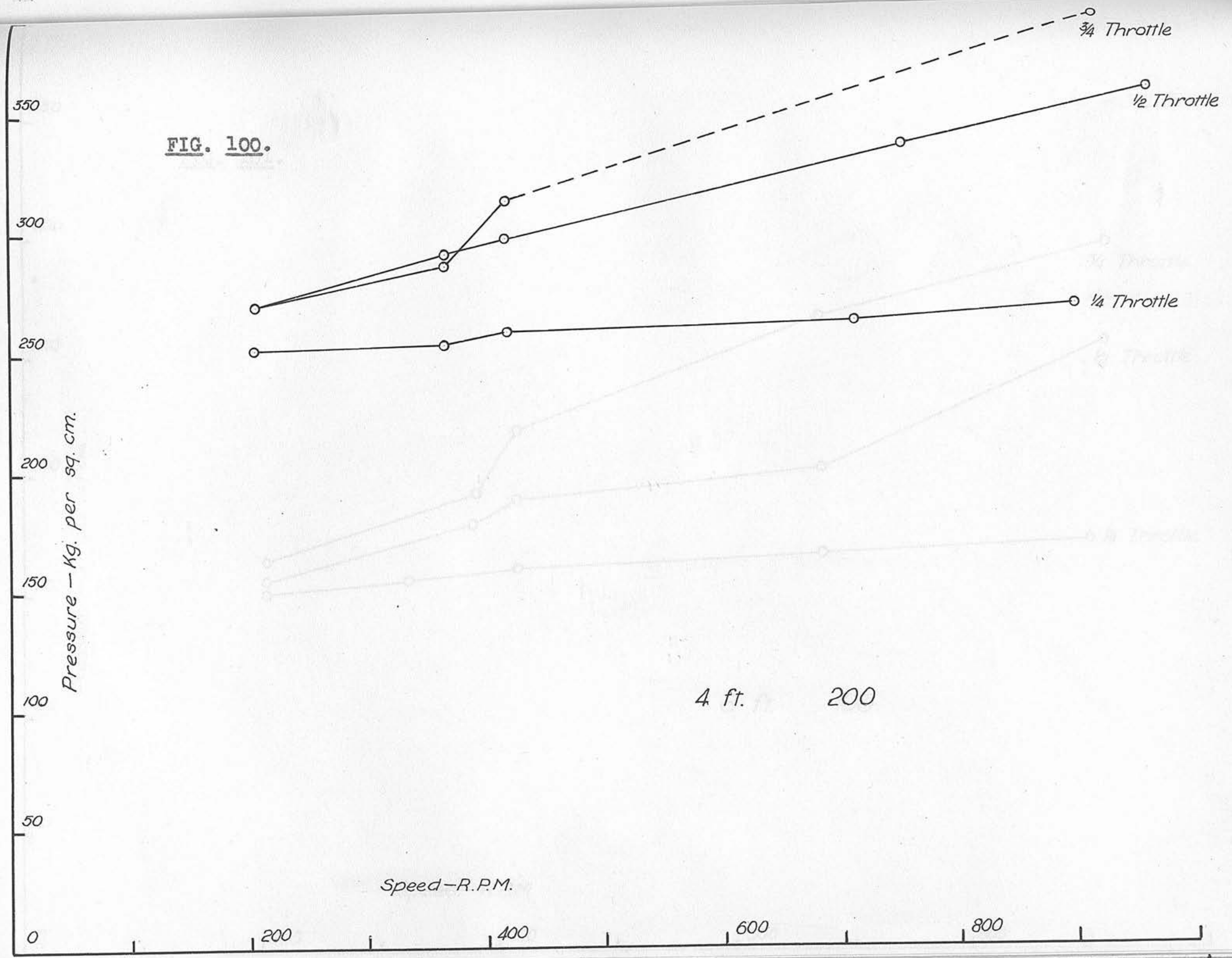
350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
¾ Throttle
½ Throttle
¼ Throttle
4 ft. 100
Speed — R.P.M.
200
400
600
800

FIG. 99.



350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
¾ Throttle
½ Throttle
¼ Throttle
4 ft.
150
Speed — R.P.M.
200
400
600
800

FIG. 100.



¾ Throttle
½ Throttle
¼ Throttle
Pressure – Kg. per sq. cm.
350
300
250
200
150
100
50
0
200
400
600
800
Speed – R.P.M.
4 ft. 200

FIG. 101.

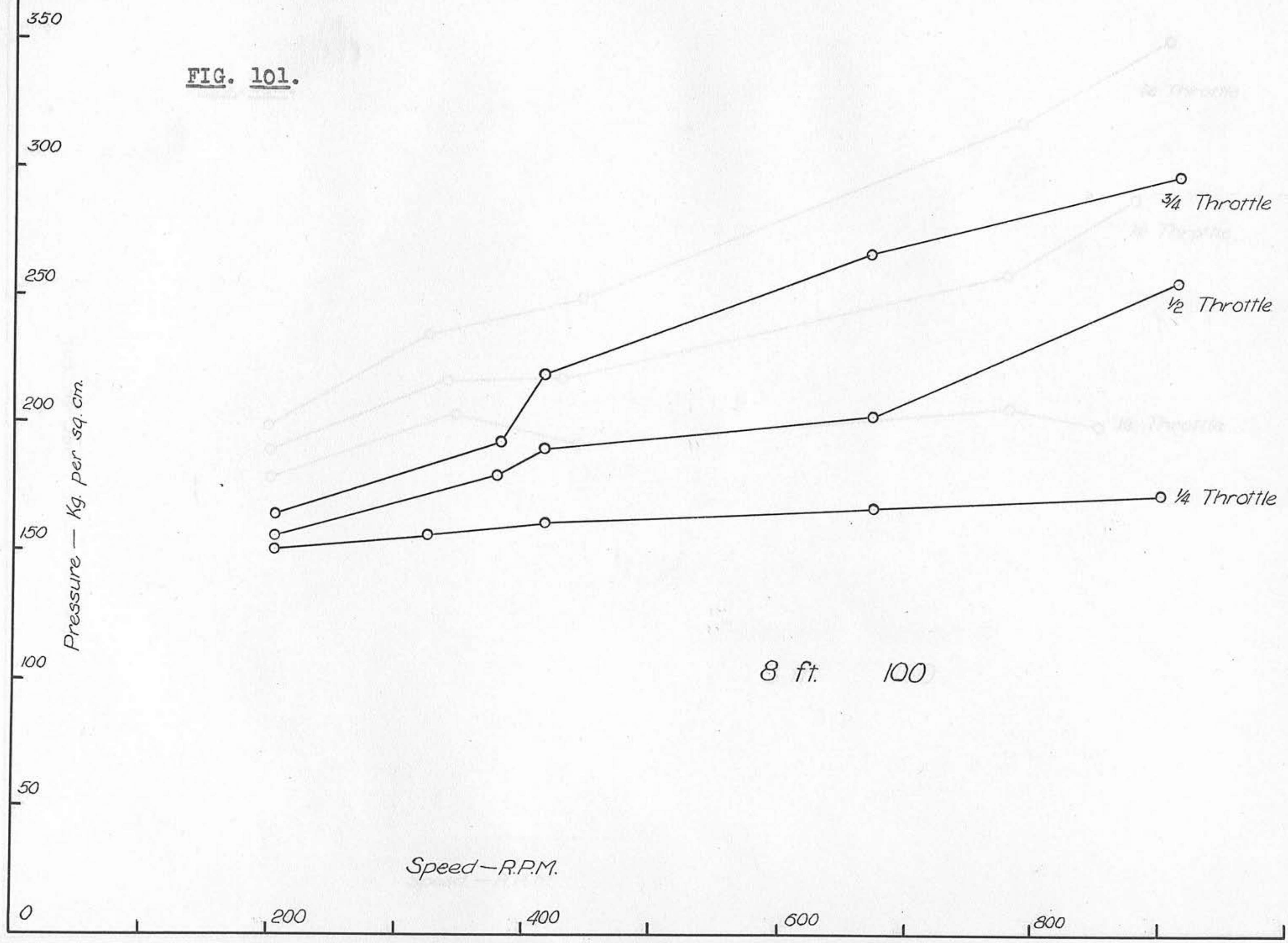

350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
3/4 Throttle
1/2 Throttle
1/4 Throttle
8 ft. 100
Speed—R.P.M.
200
400
600
800

FIG. 102.

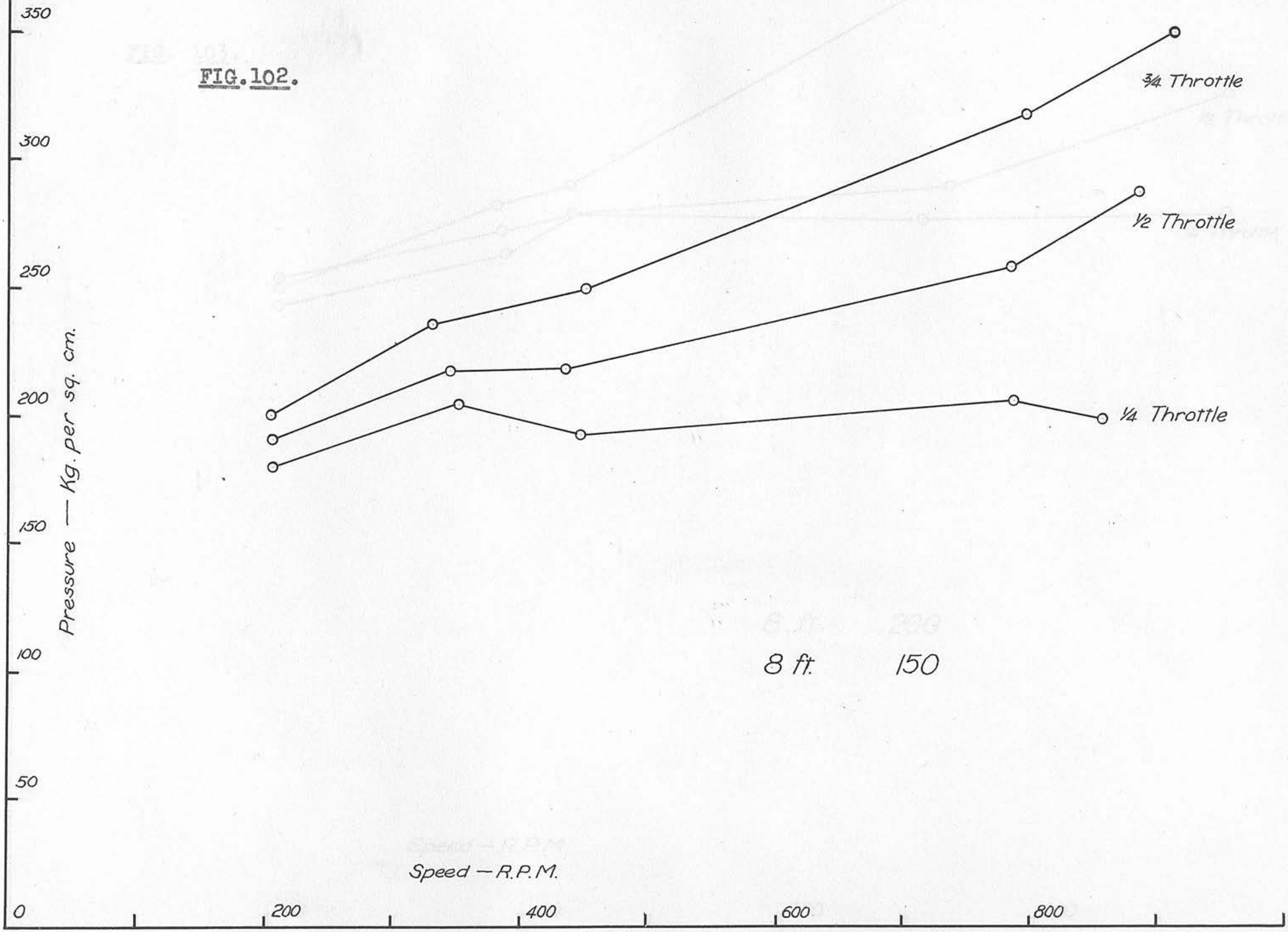
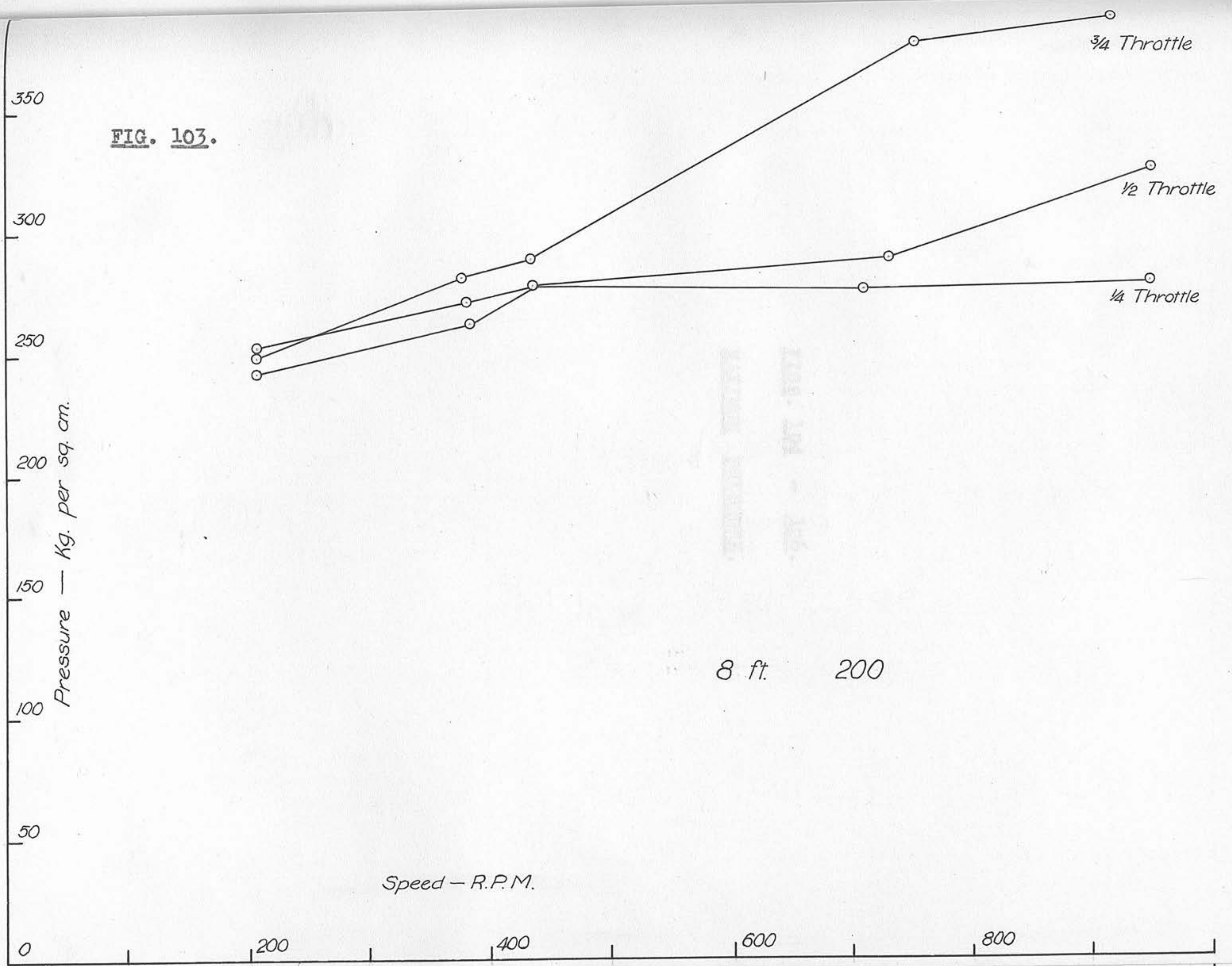

350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
200
400
600
800
Speed — R.P.M.
3/4 Throttle
1/2 Throttle
1/4 Throttle
8 ft.
150

FIG. 103.


3/4 Throttle
1/2 Throttle
1/4 Throttle
350
300
250
200
150
100
50
0
Pressure — Kg. per sq. cm.
200
400
600
800
Speed — R.P.M.
8 ft.
200

FIGS. 104 - 106.

MAXIMUM PRESSURE.

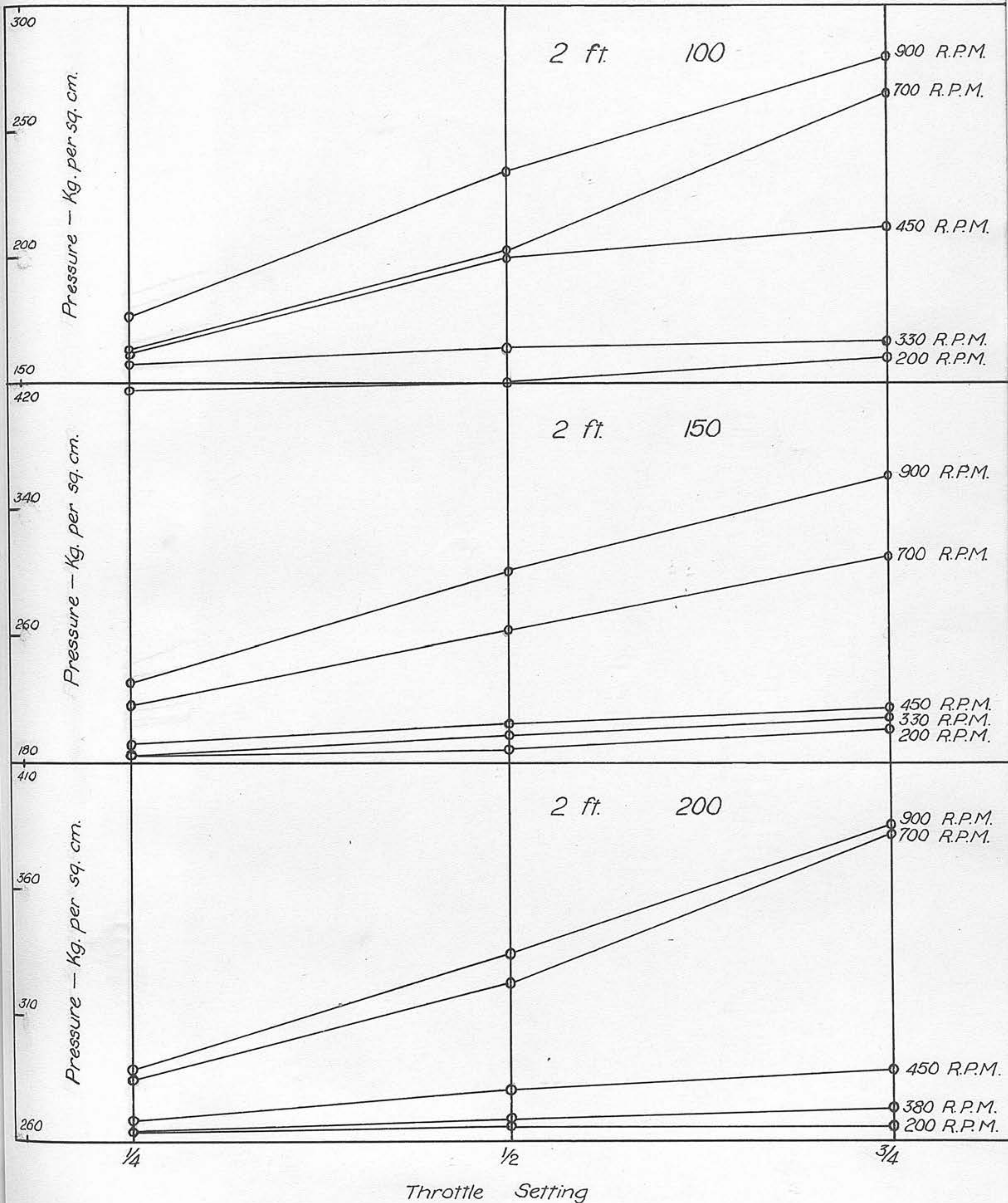

300
250
200
150
Pressure — Kg. per sq. cm.
2 ft. 100
900 R.P.M.
700 R.P.M.
450 R.P.M.
330 R.P.M.
200 R.P.M.
420
340
260
180
Pressure — Kg. per sq. cm.
2 ft. 150
900 R.P.M.
700 R.P.M.
450 R.P.M.
330 R.P.M.
200 R.P.M.
410
360
310
260
Pressure — Kg. per sq. cm.
2 ft. 200
900 R.P.M.
700 R.P.M.
450 R.P.M.
380 R.P.M.
200 R.P.M.
1/4
1/2
3/4
Throttle Setting

FIG. 104.

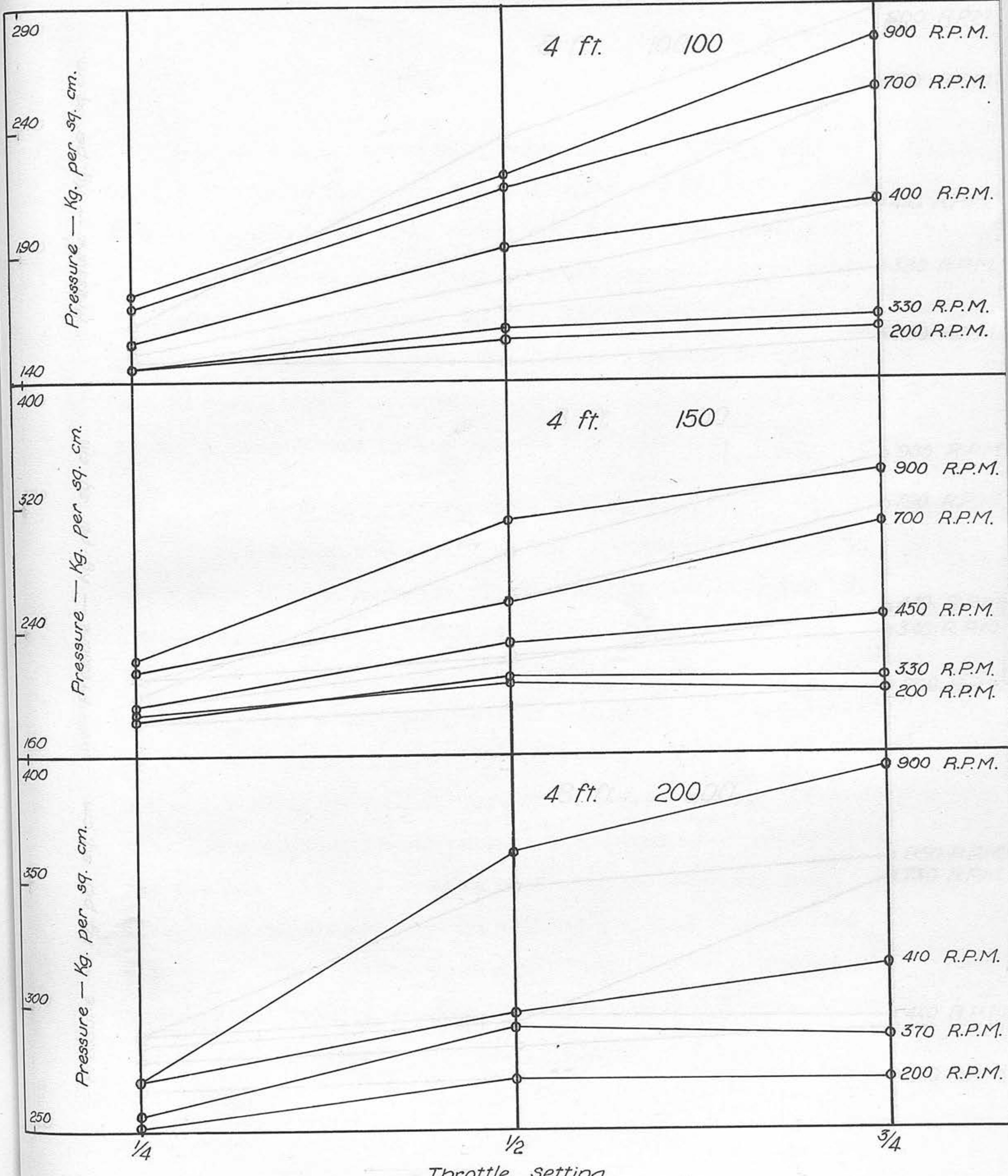

290
240
190
140
Pressure — Kg. per sq. cm.
4 ft. 100
900 R.P.M.
700 R.P.M.
400 R.P.M.
330 R.P.M.
200 R.P.M.
400
320
240
160
Pressure — Kg. per sq. cm.
4 ft. 150
900 R.P.M.
700 R.P.M.
450 R.P.M.
330 R.P.M.
200 R.P.M.
400
350
300
250
Pressure — Kg. per sq. cm.
4 ft. 200
900 R.P.M.
410 R.P.M.
370 R.P.M.
200 R.P.M.
1/4
1/2
3/4
Throttle setting

FIG. 105.

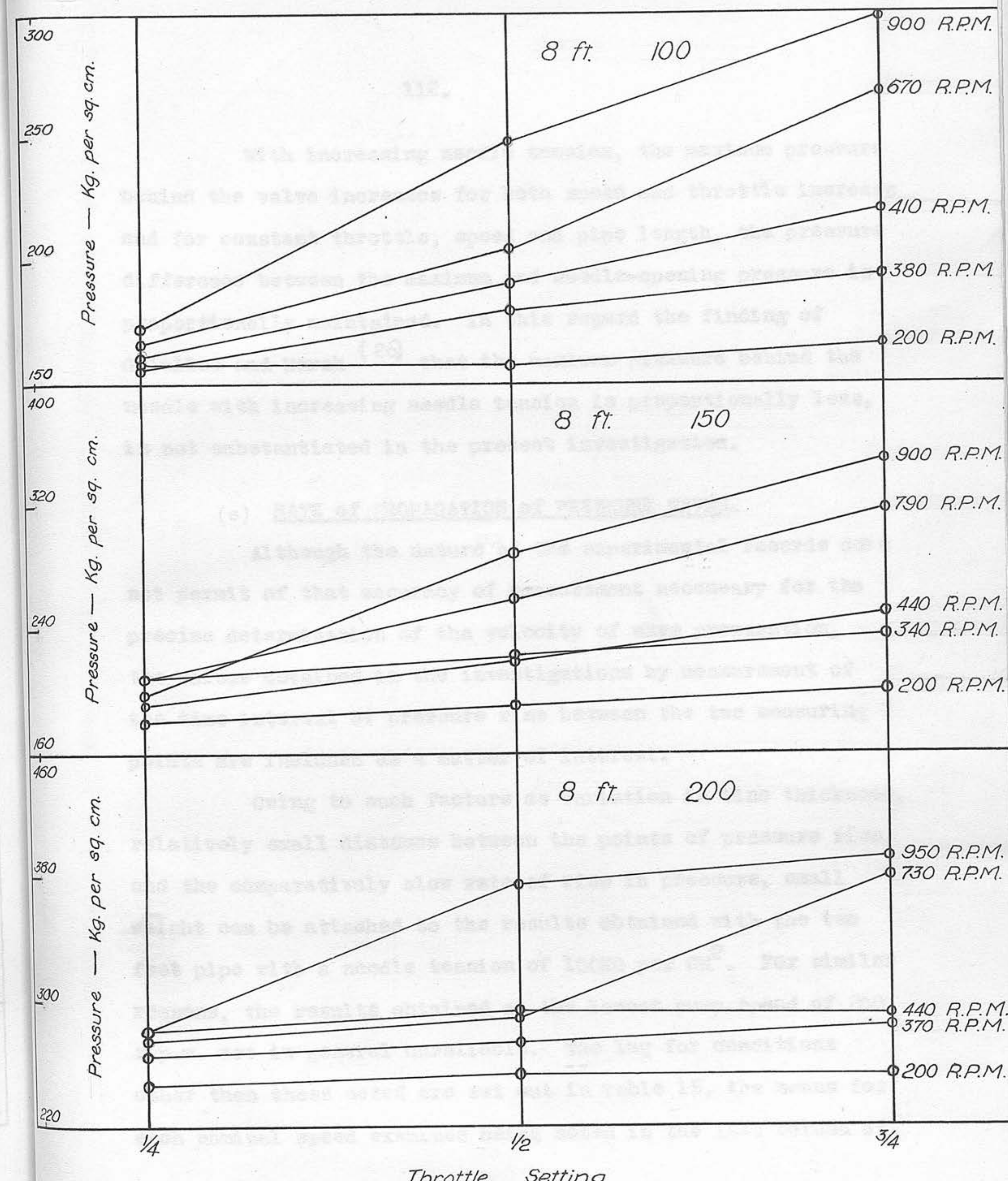

8 ft. 100
8 ft. 150
8 ft. 200
Pressure — Kg. per sq. cm.
Pressure — Kg. per sq. cm.
Pressure — Kg. per sq. cm.
300
250
200
150
400
320
240
160
460
380
300
220
900 R.P.M.
670 R.P.M.
410 R.P.M.
380 R.P.M.
200 R.P.M.
900 R.P.M.
790 R.P.M.
440 R.P.M.
340 R.P.M.
200 R.P.M.
950 R.P.M.
730 R.P.M.
440 R.P.M.
370 R.P.M.
200 R.P.M.
1/4
1/2
3/4
Throttle Setting

FIG. 106.

With increasing needle tension, the maximum pressure behind the valve increases for both speed and throttle increase and for constant throttle, speed and pipe length, the pressure difference between the maximum and needle-opening pressure is proportionally maintained. In this regard the finding of Gelalles and Marsh [26] that the maximum pressure behind the needle with increasing needle tension is proportionally less, is not substantiated in the present investigation.

(e) RATE of PROPAGATION of PRESSURE WAVES.

Although the nature of the experimental records does not permit of that accuracy of measurement necessary for the precise determination of the velocity of wave propagation, the values obtained in the investigations by measurement of the time interval of pressure rise between the two measuring points are included as a matter of interest.

Owing to such factors as variation in line thickness, relatively small distance between the points of pressure rise, and the comparatively slow rate of rise in pressure, small weight can be attached to the results obtained with the two feet pipe with a needle tension of 100KG per $CM^2$. For similar reasons, the results obtained at the lowest pump speed of 200 r.p.m. are in general unreliable. The lag for conditions other than those noted are set out in Table 15, the means for each nominal speed examined being noted in the last column of

TABLE 15.

VELOCITY of PRESSURE WAVE PROPAGATION.

FEET per SECOND.

| PIPE LENGTH | NEEDLE TENSION KG / $CM^2$ | SPEED (NOMINAL) 900 R.P.M. | 700 R.P.M. | 450 R.P.M. | 330 R.P.M. | 200 R.P.M. | MEAN VELOCITY |
|---|---|---|---|---|---|---|---|
| TWO FEET. | 150 | 5142 | 4690 | 4501 | 4616 | - | 4712 |
| | 200 | 5008 | 4795 | 4902 | 4152 | - | 4714 |
| FOUR FEET. | 100 | 4694 | 4365 | 5070 | 4245 | - | 4594 |
| | 150 | 5112 | 4700 | 4651 | 4656 | 4377 | 4700 |
| | 200 | 4783 | 4498 | 4769 | 5074 | - | 4781 |
| EIGHT FEET | 100 | 4426 | 4248 | 4220 | 4274 | - | 4292 |
| | 50 | 4512 | 4499 | 4545 | 4278 | 4371 | 4441 |
| | 200 | 4445 | 4973 | 4475 | 4583 | 4149 | 4525 |

MEAN VELOCITY:

FEET PER SECOND. 4595.

the table. Apart from the eight feet pipe, which shows consistently low values, the results are reasonably uniform in magnitude. The mean velocity of propagation derived from the mean values corresponding to nominal speeds is 4595 feet per second. Taking the bulk modulus of the fuel as 266,500 pounds per square inch, the theoretical velocity of propagation calculated from the relationship $V = \sqrt{\frac{Kg}{p}}$ , is approximately 4800 feet per second - a value slightly greater than the mean obtained experimentally.

## PART V. CONCLUSION.

The Thesis describes the design and operation of a piezo-electric indicator with associated photographic equipment and its application to the recording of certain characteristics of a Bosch Injection System.

The Injection Process is explained on a basis of wave transmission of pressure and discharge characteristics in association with needle-valve lift are examined throughout a fairly wide range of operating conditions.

Important conclusions to be drawn from the investigations may be briefly summarised as follows;

(I) Discharge increases with in-crease in pump speed and decreases with increasing needle tension, the decrease being relatively greater with longer pipe length.

(2) Initiation and termination of discharge vary with pump speed, pipe length and needle tension, the former being largely affected by the residual pressure in the fuel line.

(3) Residual pressures fluctuate throughout the load and speed range examined and are of relatively high value with longer pipes at lower speeds.

(4) Injection occurs under pressures varying with pump speed and needle tension but relatively unaffected

by pipe length.

(5) Injection at the lower speeds is characterised by irregular discharge under comparatively low pressures. With increasing pump speed, discharge occurs with increasing regularity under higher pressures and consequent improvement in atomisation.

(6) Post-Injection phenomena were not observed under the conditions investigated with normal delivery-valve setting.

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# THE ORIGIN and DEVELOPMENT

# of the

# HEAVY OIL ENGINE.

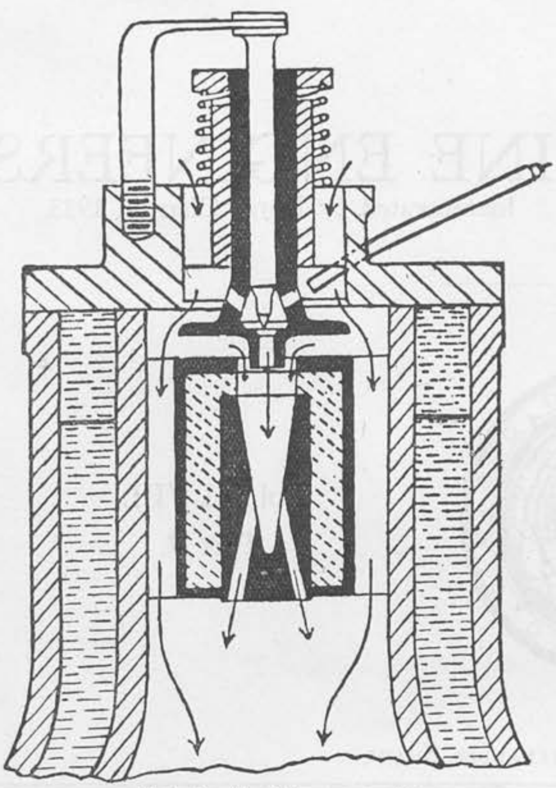

FIG. 2.—Capitaine vaporiser.

The combustible mixture was not obtained until well towards the end of the compression stroke. As shown in Fig. 3, air was drawn in during the suction stroke through the automatic air inlet valve situated at the end of the vaporiser, and compressed during the return stroke. At or near the end of compression, the fuel oil was injected in a fine spray through a suitable nozzle, and coming into contact with the heavily ribbed vaporiser was immediately vaporised and formed, with the compressed air, a

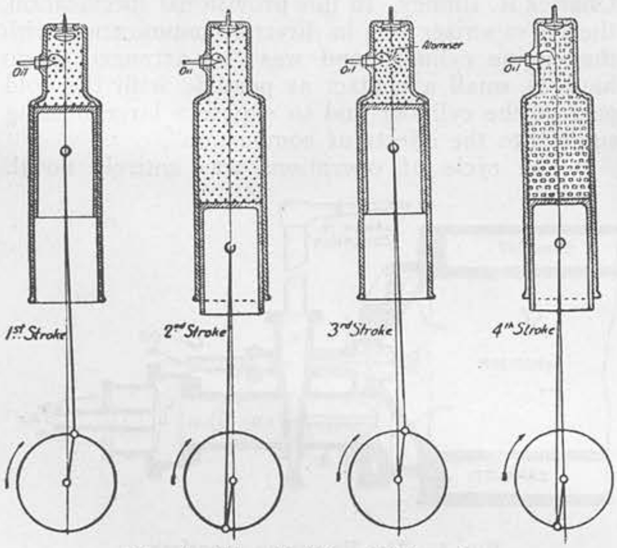


FIG. 3.—Stuart's May, 1890 cycle.

combustible mixture. The temperature of the air heated partly by compression but mostly by the heat radiated from the vaporiser was sufficiently high to cause ignition of the mixture. Stuart stated: "In the meantime the pump 'm' will have been drawing from the tank 'a' a charge of oil and at about the time that the piston commences to make its second outward stroke this oil will be sprayed into the vaporiser and with the compressed air therein form an explosive mixture which, *owing to the heat of the vaporiser,* is ignited, thereby driving the piston forward".

There was no control of the rate of combustion and detonation was the predominating feature of the cycle. There is no evidence of any engine working satisfactorily in strict accordance with the May, 1890, specification; for while *Professor Robinson describes tests which he carried out on

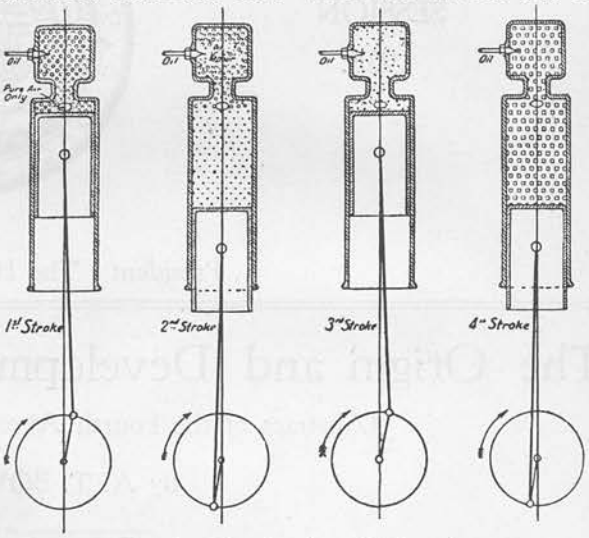


FIG. 4.—The October, 1890 cycle.

the so-called May, 1890, type of engine, this was actually (with the exception of the clearance) an engine built according to the later October patent described hereafter.

The detonating characteristics of this engine led Stuart to commence work on a modification of his first patent and before the complete specification following his May, 1890 provisional specification was filed on 5th November, 1890 he had made application on 8th October, 1890 for a modified type of vaporiser, associated with radical changes in the admission of both oil fuel and air to the engine. This type of vaporiser was shown in his complete specification for his first patent.

The cycle of operation for this October, 1890 application is shown in Fig. 4. The vaporiser was connected to the cylinder by a narrow neck, "in order", as his assistant—James Holloway—has said, "to keep the heat in the vaporiser".

With the inlet air valve altered so as to give direct communication with the cylinder instead of

*Journal Royal Society of Arts,* 1891: and "Oil Engines of Akroyd Type".

this valve being placed at the end of the vaporiser, it became possible to inject the oil into the vaporiser at practically any point of the suction or compression stroke. "When liquid hydrocarbon is employed the injection may be timed to occur at the beginning or at any portion of the suction stroke or during the compression stroke. We prefer, however, to give sufficient time for complete vaporisation in order to obtain the maximum economy of combustible liquid and this can be accomplished by injecting the hydrocarbon during the suction stroke". It is seen from the brief description given above that the second or October patent embraced the first or May, 1890 patent. The contracted-neck vaporiser and valve disposition of the later patent were not embodied in any working engine, built on Stuart's patents, of which any record is known.

Without in any way detracting from the very fine work of Akroyd Stuart, close perusal of his work and records indicates that he did not realise the far-reaching possibilities of his inventions. There is nothing to show that he conceived anything more than a very novel and efficient vaporiser, although from the very nature of his patents the logical development would have been in the nature of increased compression temperatures.

The first two engines made and numbered 101 and 102 respectively were despatched from the works at Grantham on 8th July, 1892, to the Pumping Station at Fenny Stratford. The engines were rated at 7 nominal horse power and were the only two engines made under this licence to be sold under this classification—a relic of steam engine practice. The engines operated continuously until 1923, when No. 101 was sold and to the best of the Author's knowledge it is still working, or if not, has only very recently been taken out of service. In either case, opportunity should not be lost by British engineers of preserving for all time a veritable "old man of the oil engine world", and the Author hopes that steps will be taken to preserve it in the South Kensington Museum.

In Table I a few details are given of the early 1894 type oil engines, supplied by courtesy of Messrs. Ruston & Hornsby, Grantham, while Fig. 5 shows the indicator cards taken from the 18 h.p. and 50 h.p. 1900 type.

TABLE I.

*Details of 2½ b.h.p. oil engine, 1894 type. Hornsby-Akroyd.*

| | |
|---|---|
| Cyl. dia. ... ... ... ... | 6½in. |
| Stroke ... ... ... ... | 10in. |
| Max. load ... ... ... ... | 2·75 b.h.p. |
| Mechanical efficiency ... ... | 77·0 per cent. |
| Compression ... ... ... ... | 54 to 56lb. per sq. in. |
| Crankshaft diameter ... ... | 2in. |
| Flywheel ... ... ... ... | 3ft. dia. |
| R.p.m. ... ... ... ... ... | 250 |
| Piston rings ... ... ... ... | 3 in number |
| Engine weight ... ... ... | 9cwt. |
| Flywheel weight ... ... ... | 4cwt. 1qr. |

It is a far cry from the vaporiser of Akroyd Stuart to the compression-ignition engine of the present day, but the decisive step towards it was taken when Dr. Rudolph Diesel in 1892 filed his patent (British No. 7241), for what he later termed "a rational heat motor". The chief point of interest in this rather Utopian specification is the idea of raising the temperature of the air in the cylinder beyond the ignition temperature of the fuel by the simple process of compression within the cylinder itself. It is unnecessary to describe the cycle, which is so well known and which is now of purely academic interest. Suffice it to say that the engine was designed to use powdered coal as fuel and to work according to the Carnot cycle. Only after several years of discouraging experiments and working on a second patent of Diesel's (British Spec. No. 4243) dealing with the injection of the fuel by blast air, did the M.A.N. Co. succeed in 1897 in producing an engine operating on the now well-known Diesel cycle.

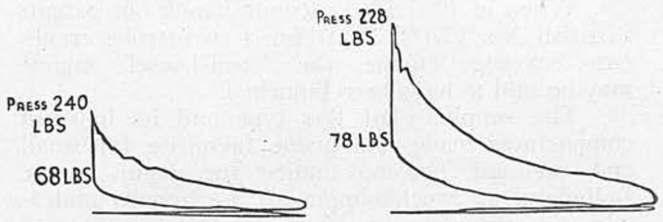


FIG. 5.—Indicator diagrams, 1900 type.

In the year 1905 when the Stuart patents expired, active competition in oil engine development may be said to have been initiated. The conflict has been waged with but slight interruption on two distinct points—firstly on the argument of mechanical versus air injection, and secondly on the principle of the four-stroke versus the two-stroke cycle.

The former question has in a natural way been thrashed out in the countries of its origin, British practice tending towards mechanical injection, and Continental towards injection by blast air. The solution of this problem has necessitated the combination of old and tried principles with the new-born ideas and principles of the true process of combustion and the almost uncanny accuracy of the modern machine tool. To say that it has been solved is to state a truism, for there is no doubt that mechanical injection has superseded injection by air, which process can now only be looked upon as a one-time convenient method of overcoming the difficult but logical means of introducing liquid fuel into the air for its combustion.

British engineers were not slow to combine the best features of the Stuart and Diesel principles, and the short history of the development of the compression-ignition engine has resulted in the gradual increase of compression pressures and consequent decrease in size and influence of the vaporiser. This is well exemplified in Figs. 6($a$) and 6($b$), which show the trend in development over the short period of six years.

When in 1903 E. A. Rundlöf took out patents (British No. 12709, 1903) for a two-stroke crank-case scavenge engine, the "semi-Diesel" engine may be said to have been launched.

The simplicity of this type and its inherent compactness made it a prime favourite for small and medium powered units for many years. Bolinders, of Stockholm, were the first to undertake the manufacture of this particular type of engine and until the expiry of the patent rights it was generally known as the Bolinder engine. It did not receive a great deal of recognition outside the Continent, although it was eminently suitable for auxiliaries of small sailing craft and indeed it was used for this purpose with considerable success.

Immediately after the War, a number of leading British firms adopted the design of this type of engine and it served a very useful purpose in bridging the gap between the vaporiser type and the Diesel proper. There is, however, no justification for its being saddled with the ambiguous name of semi-Diesel, for it has no connection with the Diesel principle. Fig. 7 shows a typical product of this period of development. The pressure in this type reaches from 150 - 220lb. per sq. inch.

Coincident with the development of the "semi-Diesel", a new type of oil engine which has been variously called "cold starting", "sprayer", "airless injection", "mechanical injection" and "high compression", was gradually introduced. From a multitude of only partly understood ideas and experiments this engine has eventually taken the form of the modern compression-ignition engine.

The evolution of the type is somewhat unique in that it has developed unheralded by any outstanding invention. On the contrary, it may best be described as the product of the collective endeavours of the leading British engineering firms, who, although working independently, have evolved a typically British engine. The principal advantage offered by this type of engine was that, by direct injection, it was possible to obtain comparable Diesel performance without the disadvantage of a compressor. In the earlier stages of development the air was compressed to a pressure of 450lb. per sq. inch and it was soon found that by due regard to considerations such as turbulence and combustion chamber design, this figure could be reduced to as low as 250lb. per sq. inch. From Fig. 6(*b*), which is typical of the construction of

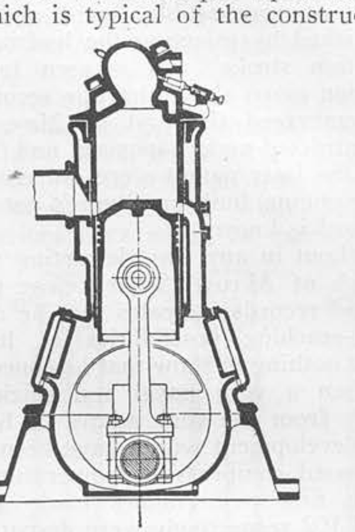

FIG. 7.—Typical two-stroke, 1915-28.

the solid-injection engine, it will be noted that while the vaporiser or hot bulb of its predecessors has disappeared, the peculiar construction of the combustion chamber ensures that there is a "hot-spot" near its geometrical centre in which the temperature of the air on compression will be little affected by the cooling water in the jackets or cylinder head. This "hot-spot", or its equivalent, is an essential element in the design of this class of engine and according to the particular combination of "hot-spot", turbulence, and formation of the working mixture, the pressure necessary for proper ignition may, as stated previously, vary between 250lb. and 450lb. per sq. inch. Combustion is mostly at constant volume, but as the injection continues well after top dead centre a greater or lesser portion of the heat liberation takes place at constant pressure. These engines, in common with the

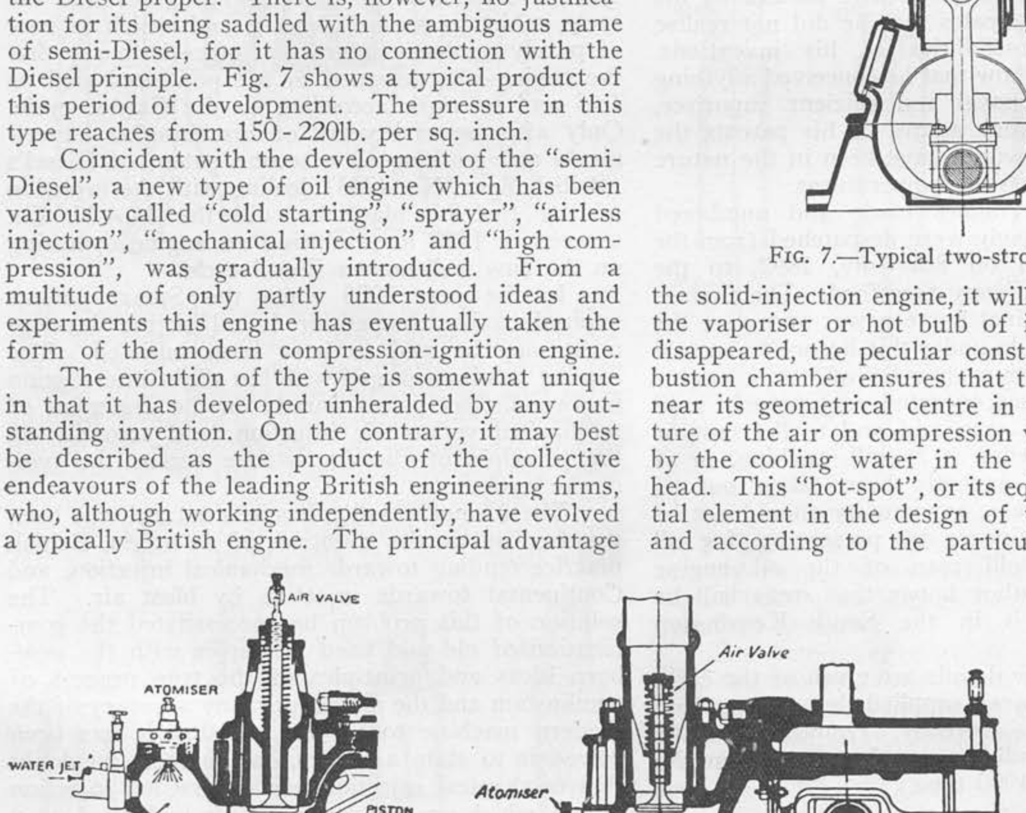


FIG. 6.—(*a*) Typical Ruston engine, 1909; (*b*) Typical Ruston engine, 1915.

types previously discussed, were essentially of the slow or medium speed types. Their design, based on marine or stationary engine practice, was essentially heavy. It was inevitable that designers would not be content to leave the realms of high speed to the petrol engine.

The main difficulties to be overcome in the production of a high-speed heavy-oil engine were in respect of injection, combustion, and weight.

## Injection.

Apart from the outstanding service of the late Sir James McKechnie, whose common-rail system was first successfully applied to submarine engines during the War, the injection problem has been solved chiefly through fine craftsmanship in the design of the pump and through a more exact knowledge of the principles of combustion within the cylinder.

The early difficulties of direct injection are appreciated when it is considered that a modern fuel pump may be required to meter 0·001 cubic inches of oil and deliver it at a pressure of 2,000lb. per sq. inch in millions of minute globules in a

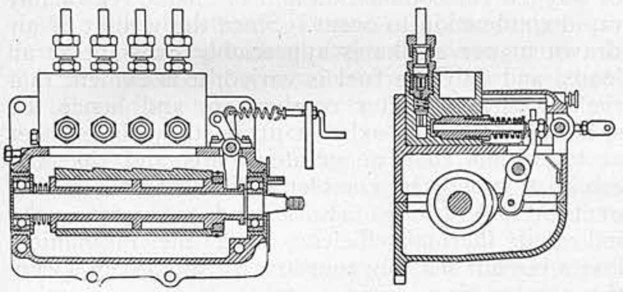

Fig. 8.—Deutz fuel pump.

fraction of one-hundredth of a second, and combined with this watch-like delicacy of movement the system must possess the ruggedness akin to that of a ship's windlass.

When compression and injection pressures were low, pump design followed the Akroyd Stuart pattern wherein the charge delivered by the cam or crank operated plunger was controlled by spill valves operated directly by the governor. With increase of speeds, however, more sensitive methods have been introduced, and following upon the work of Mr. Alan E. L. Chorlton in connection with the design of the Beardmore-Tornado engines as used in the digirible R. 101, and the subsequent work by Bosch and others, the principles of fuel injection control have now been firmly established. With numerous variations the system of control adopted with modern high-speed oil engines may be divided into two main classes: (*a*) variable pump plunger travel, and (*b*) by-passing of oil to the suction side at a point during delivery, the point being variable and under control. Regarding the first method the two principal schemes of operation are the inclined cam and the wedge. In each scheme the cam or wedge is coupled directly or indirectly to the governor.

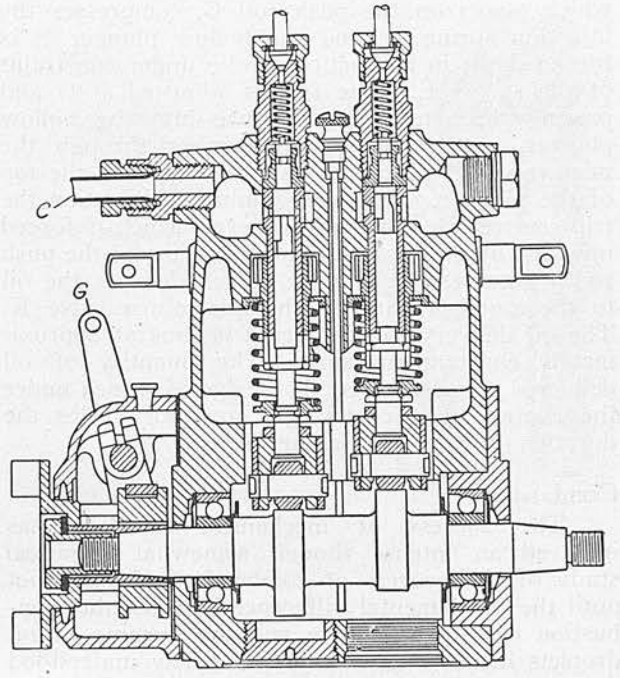


Fig. 9.—Bosch fuel pump.

The Deutz fuel pump (Fig. 8) shows clearly the former method, and (Fig. 9) showing a section of the Bosch pump, the action of which has become classical, illustrates the second method of control. A novel and very successful type of pump fitted to the Ganz-Jendrassik engine embraces a cam-operated suction with spring delivery. Referring to Fig. 10, which shows a section through the pump, the cam A, on depressing the rocking lever B,

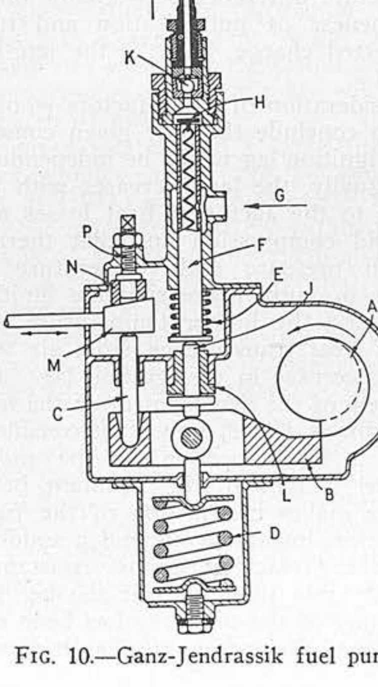


Fig. 10.—Ganz-Jendrassik fuel pump.

which pivots on the push rod C, compresses the injection spring D, and the hollow plunger F is forced down in its suction stroke under constraint of the spring E. The fuel is admitted at G and passing through a small orifice into the hollow plunger, enters the pump chamber through the inlet valve H, which makes a seating with the top of the plunger. At the beginning of injection the trip cam releases the rocking lever, which is forced upwards by the spring D and by means of the push rod J actuates the plunger, which delivers the oil to the atomiser through the non-return valve K. The oil delivery to the engine is thus at approximately constant pressure. The quantity of oil delivered is governed by the wedge M, which under the control of a centrifugal governor varies the duration of the suction stroke.

## Combustion.

The success of mechanical injection has involved an intense though somewhat empirical study of the process of combustion. It was not until the fundamental difference between the combustion of a gas mixture and the burning of oil droplets in air was realised and partly understood that any real advance in high-speed oil-engine design was made, and in spite of our knowledge of the subject at the present day it must be admitted that much room for improvement exists before this type of engine may be said to be fundamentally sound.

To appreciate the problem, it is essential to understand that when minute globules of oil are injected into the cylinder some time is necessary to enable them to absorb heat from the hot compressed air. This period depends principally upon three factors: (*a*) the degree of compression and hence the temperature difference between the oil and air, (*b*) the fineness of pulverisation and turbulence of the injected charge, and (*c*) the ignition point of the fuel.

A consideration of these factors would at first lead one to conclude that for given constant conditions the ignition lag would be independent of the speed. Actually, the lag decreases with increased speed, due to the fact that heat losses are lower during rapid compression and that therefore the compression pressure and temperature increase. The higher pressure decreasing the ignition point of the oil and the higher temperature increasing the rate of heat transference from air to oil, the result is a decrease in the ignition lag. From the point of view of the development of the high-speed oil engine this is a most important consideration.

After a small portion of the more finely divided fuel is ignited, the resultant increase of temperature makes combustion of the fuel in the cylinder almost instantaneous and a sudden rise in pressure taking place at almost constant volume follows. By this time, the air in the immediate neighbourhood of the oil spray has been consumed and oil injected thereafter must, as it were, search for its oxygen; the rate of combustion is lowered and the rate of pressure rise decreased. These stages are shown in Fig. 11.

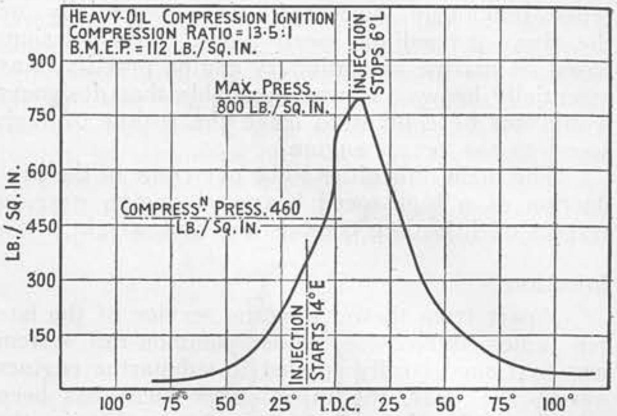


FIG. 11.—Combustion diagram of high-speed oil engine.

It will be seen, therefore, that excess air is a necessity in a high-speed compression-ignition engine in order to provide the fuel with a reasonable opportunity of obtaining an adequate supply of oxygen for combustion and to enable reasonably rapid combustion to occur. Since the weight of air drawn in per stroke is appreciably constant at all loads, and only the fuel is varied, it is evident that the temperature after combustion, and hence the specific heat of the exhaust products, will be lower at low loads than at greater loads and that the combustion will be completed in a shorter period of time. As these factors tend to increase the indicated thermal efficiency and the mechanical losses remain sensibly constant for any given speed, the compression-ignition engine fuel consumption curve shows the characteristic that as the load is decreased, the brake thermal efficiency increases until the mechanical losses predominate and cause the brake thermal efficiency to decrease. These

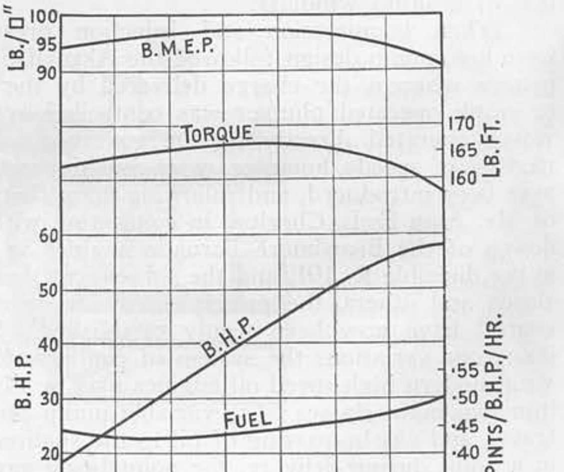


FIG. 12.—Performance curves of high-speed oil engine.

characteristics are shown in Fig. 12, which is a performance curve of a typical high-speed automobile oil engine. The constant torque characteristic of the oil engine is also well illustrated.

Desirable as these characteristics of the oil engine may be, they lead to one very great drawback—the relatively low b.m.e.p. inherent in present-day oil engine design. The greatest problem in the development of the oil engine is that of burning a rich oil-air mixture. That it has not been solved is evidenced by the fact that while the efficiency of the oil engine is higher than that of the petrol engine the b.m.e.p. is considerably less, and attempts to burn more oil with a given amount of air result in poor combustion, loss in efficiency and smoky exhaust. The multitude of combustion chamber designs, each supported by some new theory, indicate the almost frantic efforts being made to solve the problem. Controlled turbulence, bringing a greater weight of air into contact with a given quantity of atomised oil in a shorter period of time, is the guiding principle of every new type of design. While considerable success has been achieved in some well-known designs, it must be apparent to even the most casual observer that in spite of the ingenuity of some of these designs the problem is being tackled from the wrong end. In the first place, the characteristics of the fuel used in oil engines are imperfectly understood. What, for instance, is understood by the so-called ignition-point of a fuel? It is doubtful if two engineers would place the same meaning upon it. Secondly, the combustion process of such fuel has received scant analytical study. Oil engine fuel is selected more or less haphazardly, and comparatively little has been done to reduce it to specification apart from certain physical characteristics of the "not greater than" - "not less than" type.

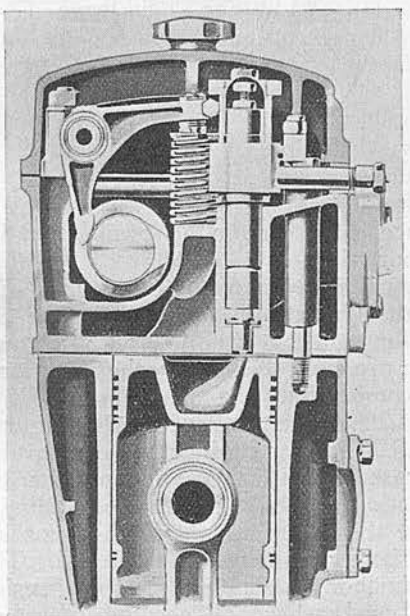

FIG. 13.—Leyland cylinder head.

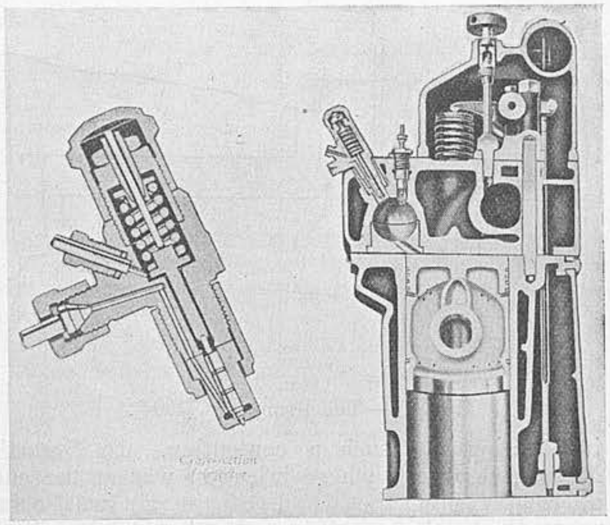

FIG. 14.—A.E.C. cylinder head.
(A.E.C. copyright).

The earliest efforts in high-speed oil-engine design took the form of increased injection pressures and very high pressures were not uncommon. Penetration falls off as the degree of atomisation increases, and, as mentioned previously, the modern tendency is to create turbulence in preference to fine atomisation, either by direct methods or by means of the so-called "pre-combustion chamber", "ante chamber" or "air cell".

The former method is exemplified in Fig. 13, which shows the combustion chamber of the Leyland six-cylinder $4\frac{5}{8}$in.$\times 5\frac{1}{2}$in. type. As will be seen from the diagram, the air is forced into a compact space situated in the piston head, which is unaffected to a large extent by any water-cooled surface and into which, as the piston nears the end of the compression stroke, the air will enter with a fair degree of swirl.

The process is carried a stage further in the A.E.C. six-cylinder engine (Fig. 14) which develops 130 b.h.p. at 2,400 r.p.m. In this case, the combustion chamber is located entirely within the cylinder head and is of spherical construction communicating with the cylinder by a tangential air passage. During compression the air is forced through this passage and enters the combustion chamber with a high degree of rotational swirl, ensuring a uniform distribution of the fuel throughout the air.

The pre-combustion chamber (with which the last mentioned type of combustion chamber should not be confused) is designed to use the partial combustion of a small portion of the fuel to ensure complete and rapid combustion of the whole. It might be considered that the pre-combustion chamber of to-day is a direct descendant from the Akroyd vaporiser, but actually when the question is considered impartially, there is no

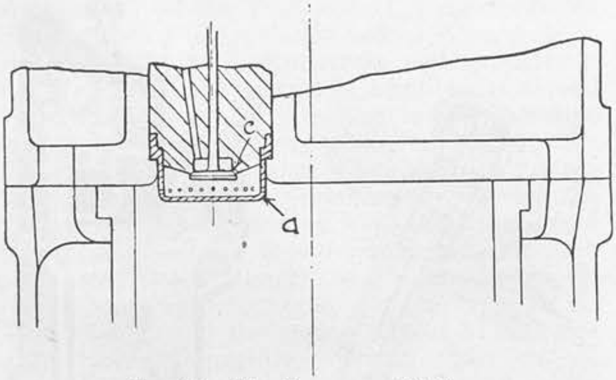


FIG. 15.—The Brons cup (1904).

justification for such a contention. In Stuart's May 1890 patent, where injection was at the end of compression, the whole of the air was compressed into the vaporiser. In the October 1890 patent, the function of the narrow neck was to make the vaporiser act as an oil-vapour isolating device. It is evident, therefore, that Stuart did not regard his vaporiser as in any way taking the place of the modern pre-combustion chamber. Technically, however, it must be looked upon as influencing the evolution of the modern pre-combustion chamber.

Similarly, the Brons patent No. 14165 of 1904 must be regarded as not anticipating but certainly influencing future design. In this patent (Fig. 15) oil is drawn during the suction stroke through the valve C into the capsule A in the cylinder head. The capsule, being provided with a number of fine holes near the bottom, isolated the vaporised oil during the suction stroke and ensured that combustion took place only towards the end of the compression stroke when, upon the hot compressed air being forced through the holes into the capsule, a small portion of the oil burnt increased the pressure within the capsule, forcing the remainder

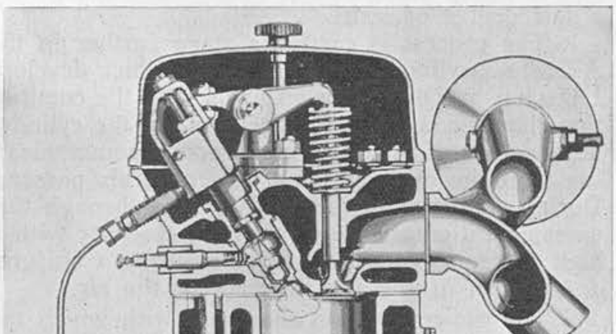

FIG. 16.—Mercedes-Benz cylinder head.

of the oil into the main cylinder with a high degree of turbulence which ensured its rapid combustion with the air contained therein.

Following the developmental work of Sabathé, Ricardo, Steinbecker, the A.E.G., Avid and Korting, L'orange by his British Patent No. 184692, 1921, may be said to have brought the idea to its present state of perfection, or perhaps it might be better to say, imperfection. The Mercedes-Benz combustion chamber shown in Fig. 16 illustrates a modern example of its class. The pre-combustion chamber communicates with the cylinder via holes in a separating grid. Following upon compression, a small portion of the oil injected burns and the remainder is forced into the main combustion cylinder with considerable

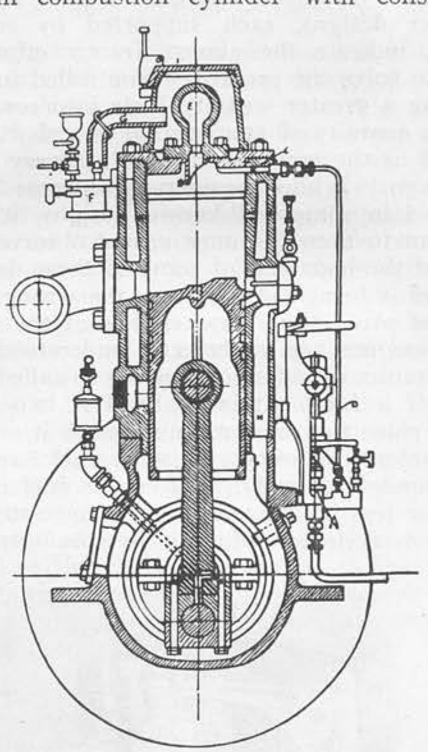

FIG. 17.—Meitz and Weiss (1900).

turbulence, thus completing combustion in a relatively short time.

The chief advantage of this system of injection over direct injection is that complete and rapid combustion can be obtained with a much lower injection pressure. Generally, the injection pressure in pre-combustion chamber designs is approximately 1,200lb. per sq. inch, while it may be as high as 5,000lb. per sq. inch for comparable results in the case where direct injection is used. Such low injection pressures eliminate dribbling troubles, and have the added advantage that nozzles having holes of larger diameter may be used. This factor has the further advantage that topped residual fuels may be used, where on account of erosion, etc., their use with direct injection is impossible. Further, as only a small portion of the fuel is burnt at constant volume, the remainder being burned under constant pressure conditions, the maximum pressure is not so high as in the case of the direct-injection system. Constant pressure combustion, coupled with the lower compression ratios usually employed in this type, have the effect, however, of lowering the efficiency and the brake m.e.p. This seems to be adversely affected to the extent of about 7 per cent. compared to the direct injection type. On account of the lower compressions obtaining with the pre-combustion chamber design, some form of starter is usually required. In most cases this consists of electrically-heated plugs.

The air chamber method of securing rapid combustion in compression-ignition engines may be said to have originated in a patent (British No. 21649) taken out by Meitz and Weiss in 1900 and shown diagrammatically in Fig. 17. It will be seen from this illustration that the combustion chamber separated from the main combustion space in the cylinder by a narrow neck terminates in a lip. Towards the end of the compression stroke fuel is injected through the jet which impinges on this lip, and is partly deflected towards the narrow neck in the combustion chamber E. That portion entering the hot bulb becomes ignited and the subsequent rise in pressure drives the unburnt oil, products of combustion, and air into the main cylinder with a high degree of turbulence ensuring a rapid dispersion of the unburnt oil through the air in the main combustion chamber.

The modern version of the air chamber or air cell is due to Franz Lang, of Munich, whose patent was taken over by the Aero Motor Company of Kuersnacht and later passed into the control of the Robert Bosch Company, which issues licences for its manufacture. The original design has suffered considerable alteration and modification in the numerous British and Continental engines to which it has been fitted. The air cell consists of three parts :—

(1) The air storage chamber;
(2) the venturi;
(3) air space in cylinder head, which is generally kept down to a minimum.

The nozzle is set to inject the atomised fuel directly into the venturi and the ignition lag is sufficient to ensure that combustion originates near the throat, causing combustion of the air within the venturi at constant volume. As the piston descends the air within the storage chamber issues through the venturi and is burnt under approximately constant pressure conditions.

Having now traced the development of the oil engine and especially what may be called the logical supremacy of airless injection over air injection, it remains to consider the extent to which the oil engine has invaded those fields which had previously been held by the steam or petrol engine. For this purpose, it is proposed to consider its modern application to transport by road, sea, rail and air, a short section being devoted to electricity supply.

## Road Transport.

From previous considerations dealing with the development of the high-speed oil engine, it will be apparent that, due to a relatively low b.m.e.p. and relatively high maximum pressure, the oil engine suffers the initial disadvantage that for equal power characteristics it is necessarily heavier than a petrol engine. This difference in weight, however, with the advanced design of the modern oil engine is not excessive for commercial vehicles. From an analysis of data relating to well-known engines the average weight per b.h.p. of the modern oil engine as applied to road transport is about 16lb.

It follows that the initial cost must be greater, and although the figure varies, it may, for the usual type of auto-oil engine, be taken as about 30 per cent. more than for the corresponding petrol engine. There are, however, many compensations offsetting these initial disadvantages. The high brake thermal efficiency of the oil engine, combined with the present low cost of oil fuel suitable for high-speed engines, make it a very attractive proposition for the commercial vehicle user; apart from a slightly greater cylinder wear and big-end bearing replacements, the maintenance of the oil engine compares favourably with that of the petrol engine.

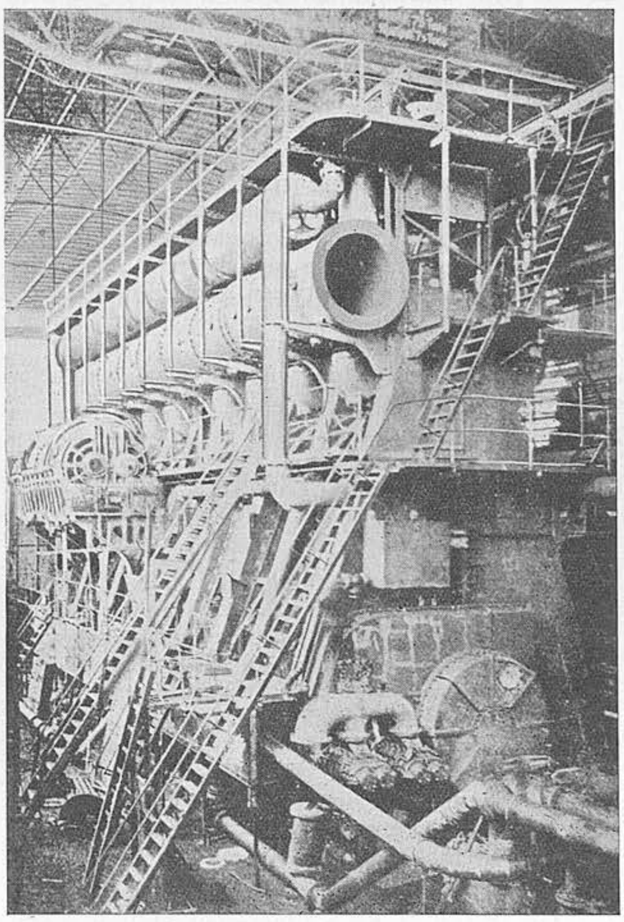

FIG. 18.—B. & W. 22,500-h.p. oil engine.

It is not certain that the cost of fuel oil suitable for high-speed oil engines will remain at its present low level. Indications are not lacking that governments look towards taxation of imported fuels, and in some cases small taxes are already operating. It is hardly conceivable, therefore, that industry will for long be able to enjoy cheap fuel, at any rate in those countries having a plentiful supply of high grade coal. The matter is entirely different where a country is forced to import both oil and coal; in that case, there is not the same political incentive to protect home industries.

There must arrive a time in the development of the oil engine when a balance between petrol and oil values will have to be struck, but on account of the inherently high efficiency of the oil engine as compared with the petrol engine and its marked economy at low and medium operating loads, even in the event of the price of the two fuels reaching the same level, there are very few commercial concerns where the saving in fuel costs on a ton-mile basis would not more than offset the interest and depreciation on the initially higher cost of the heavy-oil engine.

The first oil-engined passenger vehicle to be introduced into England, was a converted Karrier six-wheeled chassis fitted with a Mercedes-Benz engine. This was placed in commission by the Sheffield Corporation in March, 1930. Oil-engined omnibuses and trucks had been on the road in Germany since 1927, but the Sheffield omnibus marks a distinct milestone in oil-engined transport in England. Since that date the progress of the oil-engined omnibus and truck has advanced at an ever accelerating rate. Illustrating this point, Table II shows the number of compression-ignition buses operating under the Manchester Corporation Transport Department between May 28th, 1933 and May 27th, 1934.

Table II.

| Month | May 28th | June 25th | July 30th | Aug. 27th | Sept. 24th | Oct. 29th | Nov. 26th |
|---|---|---|---|---|---|---|---|
| No. of buses | 54 | 68 | 70 | 78 | 81 | 83 | 92 |

| Month | Dec. 31st | Jan. 28th | Feb. 25th | Mar. 31st | April 29th | May 27th |
|---|---|---|---|---|---|---|
| No. of buses | 88 | 90 | 94 | 121 | 128 | 138 |

During the week ending May 27th, 1934, the total mileage covered was 133,933 and the gross estimated saving (based on previous experience) of fuel and lubricating oil was 1·954 pence per mile. After making allowance for the difference in oil engine licence costs this is reduced to an estimated net saving (not including interest, depreciation, etc.) of 1·644 pence per mile. As indicating the general trend, there are now in England over 12,000 road vehicles using compression-ignition engines and it is estimated that this number will increase at the rate of over 10,000 per annum.

The design of the compression-ignition engine for motor vehicles closely follows that of the modern petrol engine. That the oil engine is in many cases interchangeable with the petrol engine of equal power on a standard chassis with the same gear box and overall gear ratios marks the degree of flexibility obtainable. The stroke-bore ratio, on account of the high compression ratio and subsequent small clearance space, is some 10 to 15 per cent. greater than the petrol engine. Aluminium alloy pistons are almost universally employed and in some cases aluminium cylinder blocks with steel or cast-iron heads have been successful. Crank shafts are usually of nickel chrome or nickel chrome molybdenum having a tensile strength of from 55 /70 tons per sq. inch. From the experience gained during the past five years, it can be safely stated that faults due to the injection system including the pump and atomisers are in no way as numerous as those experienced with the carburettor and ignition system of the petrol engine. The modern oil pump has proved a reliable and stable unit over long periods of continuous working, and it seems reasonable to suggest that it will eventually lend itself to such further refinements as will enable it to meter and deliver the small oil quantities necessary for the high-speed multi-cylinder touring car engine which must, of course, be considered the ultimate application of the high-speed oil engine.

## Central Station Electrical Generation.

Regarding the generation of electricity by oil-engines, the time has now arrived when special considerations determine the choice of prime movers. Increasing electrical system interconnection of generating stations throughout the world, the rapidly increasing demand for very large machines and the desirability of concentrating generation in fewer and fewer stations, rule out the general use of oil engines. Frequency control demands a high degree of accuracy which cannot be attained by oil-engined generators. Discussing tandem machines only, a 105,000 kW. machine is in operation at Battersea, London, and a 160,000 kW. machine has been in service several years at Hudson Avenue, New York. This trend is confirmed by recent installations in Great Britain and the data published by the Central Electricity Board, wherein it is seen that in 1934 the maximum size of machine ordered was 75,000 kW. and the average size 45,000 kW. Continental and American practice follows the same lines. In England the Electricity Commissioners' Annual Reports show the substantial decline in the number and capacity of oil-engine stations during recent years. The thermo-dynamic efficiencies of large turbo-alternators reach 35·0 per cent.

Despite this trend, oil engines are favoured by a few station engineers for peak load purposes only, but in these cases the maximum capacity has only reached 22,500 h.p. (16,800 kW.) at 115 r.p.m. This engine is of the B. & W. double acting two-cycle type having eight cylinders 33in. diam. and

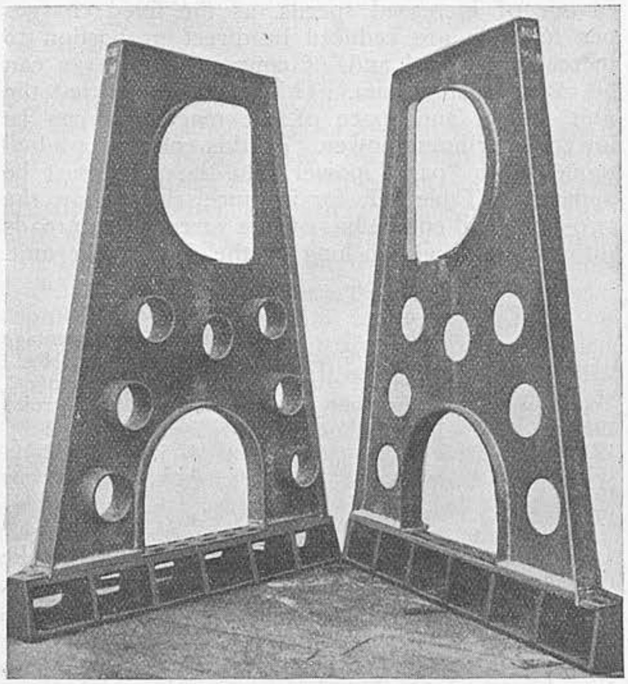

FIG. 19.

59in. stroke, the continuous normal output being 15,000 kW. The guaranteed fuel consumption is put at 0·585 - 0·580lb. per kW./hr., the lowest guarantee being 0·530lb. per kW./ hr. at 17,000 h.p. (12,700 kW.). The overall length of the engine is 66ft., 40ft. high, 27ft. 9in. wide and weight 1,200 tons. This machine is illustrated in Fig. 18.

It would seem that the future application of the oil engine for electricity generation lies in small isolated places where small public, private or factory supplies are being initiated and where fuel oil is or can be made readily available. This would appear to cover most of the non-industrial countries of the world and thus the future of the oil engine for electricity generation is fairly well defined.

Marine.

In marine work the oil engine is particularly suitable, but it is not the Author's intention to detail its development. This has already been adequately covered by technical publications devoted to the subject and by many excellent papers in the proceedings of institutions. Since the installation of the B. & W. oil engines in the "Selandia" in 1912 there has been continuous progress under the most exacting conditions. At sea, the oil engine has definitely proved that it can compete in respect of reliability with the well established steam prime mover. From economical considerations, the question of installing oil engines is largely one of the relative cost and quality of coal available and the particular run on which the ship is engaged. With reasonable oil price coupled with lower production costs attendant upon advanced design and technique, the advantage is becoming increasingly realised. Regarding the question of oil costs, it should be borne in mind that apart from purely coastal shipping, the marine oil engine is to a large extent independent of that political interference which by the imposition of excessive taxes on imported fuel may retard the development of the oil engine on land. The world distribution of oil supplies ensures a freedom of purchase to the oil-engined ship which the land user does not enjoy.

The most striking development of recent years is that of airless injection and the disappearance of air injection. As a means of establishing and improving the oil engine, one must look back upon air injection with gratitude but few will regret its passing. Supercharging of the four-stroke cycle engine, resulting in about 30 per cent. increased power over atmospheric induction with equal cylinder displacement, has almost become universal, and the reliability of supercharging, at one time considered doubtful, is now definitely established. Reduction in weight has been sought from two angles. Firstly, by a general tendency towards increased speeds and, secondly, by employing welded frame construction. The latter has been increasingly used in land and railway oil-engine installations and its adaptation to the marine engine was inevitable. Illustrating this latter tendency Figs. 19 and 20 show the application of the welded frame construction to the Doxford oil engine. The illustrations refer to the four-cylinder 2,900 b.h.p. type as installed in the motorship "Devon City". The Doxford opposed-piston type is, of course, eminently suited for welded construction on account of the fact that the frame work is required to take only the side thrust from the connecting rod, but due to the possi-

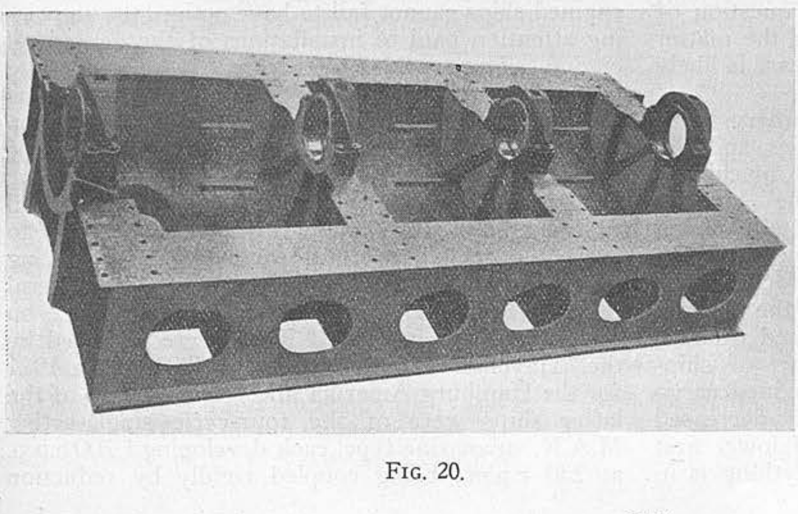

FIG. 20.

bility of the economical disposition of material and subsequent decrease in weight, together with the highly developed technique of modern welding methods, the principle undoubtedly must be extended to all types of framing. From Fig. 19 it will be seen that the main frames are composed of two H sections welded together and tied with boiler tubes welded to the frames. The column bridge and cylinder supports are also of welded construction, the complete assembly being of extremely neat and pleasing appearance. In a later construction, the bed-plate is also of welded construction. Fig. 20 shows it is built up of steel plates and girders and the bearings carried in steel shells dovetailed and welded into the cross members.

In spite of its long evolution and slow growth, the present-day marine oil engine still bears a striking resemblance to its predecessor and rival—the reciprocating steam engine. Cast-iron and cast-steel remain largely in its construction and it is in most cases directly coupled to the propeller. It is inconceivable that marine oil-engine designers can for long neglect the most recent advances in metallurgy and transmission, for lightness of modern materials allows of high speeds; modern transmission systems also permit the application of these high speeds, and these factors combined allow of reduced cost and reduced space. That high speed is not inconsistent with reliability is shown by the modern aeroplane engine which, using higher loadings than would ever be contemplated in marine practice, gives a reliability in service that might well be envied by some of its lower speed contemporaries. With its lower heat stresses and higher working pressures and temperatures, the high-speed oil engine is eminently suited for the higher loadings that must be undertaken if the oil engine is economically to take its due place in high-speed cargo vessels and to extend its activities to the larger and faster ocean liners. The high inertia stresses which become paramount in high speed work can be adequately catered for by the use of light alloys for pistons and even for connecting rods. As regards the much discussed question of cylinder wear, the evidence available on the matter would suggest that no unduly large increase is likely to occur with high speeds.

The question of the type of machinery to be installed is invariably one of economics and the development of any particular class of machinery may, generally speaking, be traced to the economic conditions obtaining or estimated to obtain during the installation and life of that machinery. With the decreasing costs of oil engine manufacture consequent upon increasing production and the increasing operating efficiencies due to advanced designs, there is a distinct tendency on the part of shipowners to increase the speed of oil-engined cargo vessels. Given the two factors of decreased interest and depreciation as a result of lower first costs and increased fuel economy, everything is in favour of increased speeds, as the fixed charges per ton-mile are reduced in direct proportion to increase of speed and, of course, more cargo can be carried per annum. This is providing that the unit weight and space of the machinery can be lowered per horse power. To this end, the gradual tendency to "pack" power into the ship must be apparent. Table III, for instance, shows how the two-stroke oil engine has of late years made inroads into the field held so long by the four-stroke unit.

TABLE III.

| Year. | Total oil-engined ships, 1,000 tons and over. | Percentage four-stroke. | Percentage single-acting two-stroke. | Percentage double-acting two-stroke. |
|---|---|---|---|---|
| 1929 ... | 181 | 63·5 | 29·3 | 7·2 |
| 1930 ... | 240 | 53·3 | 42·5 | 4·2 |
| 1931 ... | 176 | 51·8 | 42·1 | 6·0 |
| 1932 ... | 50 | 58 | 26 | 16 |
| 1933 ... | 63 | 35 | 46 | 19 |

Even with the double-acting two-stroke developed to its fullest potentiality on present lines, the time will arrive when the limits of usefulness imposed by space, stress, and temperature considerations will prevent further development. Simply to state that by increased speed these obstacles can be overcome, is to ignore the fundamental difficulties involved. Firstly, as the speed of the oil engine is increased it becomes increasingly sensitive to the quality of the fuel it uses, and with our present knowledge of airless injection it is doubtful if speeds exceeding 600 r.p.m. could be usefully employed with the utilization of existing low priced oil fuels. Secondly, it is very doubtful if propeller design could be successfully accommodated to the changed conditions of such speeds. Already, in anticipation of the dead-end of slow speed production, several designs embracing new thought on the question have been developed. Especially is this noticeable on the Continent, where shipowners have taken serious interest in the oil engine and where every encouragement is given to stimulating new ideas in design.

The careful observer of tendencies in oil-engined ships cannot fail to have noticed the increasing attention paid to installations of the geared oil engine. The troubles and fears of the immediate post-war period of geared-turbine installations having been overcome by better machine shop practice and by a clearer insight into the problem of torsional oscillation, the way has been opened for the adoption of the higher speed geared marine oil engine, and already many installations testify to the economy and reliability of the system. Among the earliest geared oil engine drives were several paddle-wheel tow boats constructed for service on the river Volga in 1909. These were followed by the "Havelland" and "Munsterland", built in 1921 for the Hamburg-Amerika line. The engines of the latter ships were of the four-cycle single-acting M.A.N. submarine type, each developing 1,700 b.h.p. at 230 r.p.m., being coupled rigidly by reduction gearing to the propeller shafts. A gear ratio of 1 : 2·7 was employed, giving a propeller speed of 85 r.p.m.

During 1924-25 the motorships "Monte Sarmento" and "Monte Olivia" were completed for the Hamburg Sud-Amerika Line and were followed later by the "Monte Rosa" and "Monte Pascoal". In each of these ships geared oil engines were installed and after ten years' service the installations are still giving very satisfactory performance.

The two older ships were fitted with four six-cylinder single-acting supercharged four-stroke engines built under licence from the M.A.N. by Messrs. Blohm and Voss. Each engine produced 1,500 b.h.p. at 215 r.p.m. In each case two engines were rigidly coupled to each propeller through gears giving a speed reduction of 2·8 : 1. The gears were of the herring-bone type with 45° angle and 0·816 circular pitch. The pinion was 32in. diameter (124 teeth), the larger wheel being 90in. diameter (348 teeth). The later ships were unsupercharged, but developed the same b.h.p. by increasing the cylinder diameter from $23\frac{5}{8}$ins. to $24\frac{3}{16}$in. The stroke was $27\frac{9}{16}$in. in each case.

With a view to eliminating shock and torque variations to the gears as well as for ease in manœuvring, there has been an increasing tendency in geared installations to insert a flexible drive between the engine and the gears. While the necessity of flexible couplings is debatable, there is no doubt that the practice has done much to conceal any inherent deficiencies in the system and to create confidence which might otherwise be lacking.

The most usual form of flexible connection is of the hydraulic type, typified by the Vulcan hydraulic coupling. This coupling, which is based on the design of the well-known Föttinger coupling and is used successfully on many Bauer-Wach installations, not only relieves the gears of shock but permits wide variations in propeller speed while maintaining constant engine speed. For manœuvring purposes, this is a great asset. In tug work, for instance, since the slip of the coupling at low speeds is high, the tug can be brought to the tow comparatively slowly and evenly. A marine type of coupling, where means are available for varying the quantity of oil in circulation enabling any required slip to be made between the engine and the propeller for any given engine speed, is shown in Fig. 21.

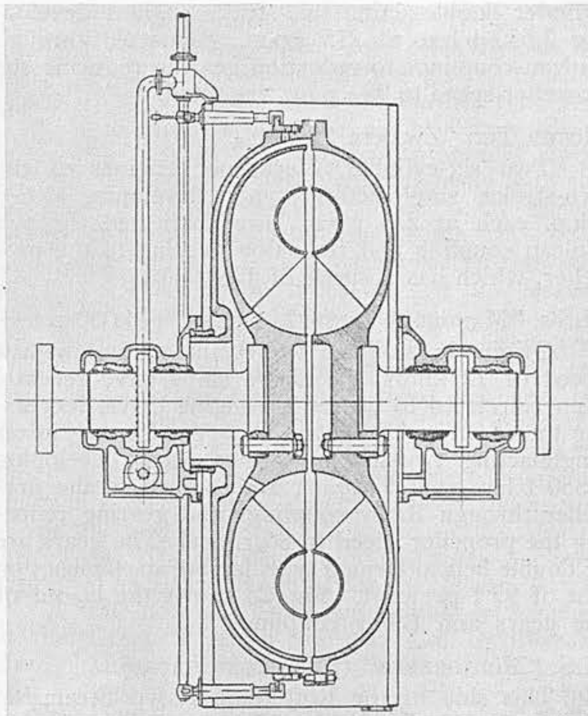

Fig. 21.—Hydraulic coupling—marine type.

In order to show the widely divergent use of the geared oil engine drive some details of recent installations are recorded below.

### Twin-screw M.S.'s "St. Louis" and "Milwaukee" (17,000 tons gross).

Each ship is fitted with four six-cylinder M.A.N. double-acting two-stroke engines, each developing 3,000 b.h.p. at 220 r.p.m., two engines being coupled to each propeller shaft through gearing, giving a propeller speed of 110 r.p.m. The "St. Louis" is fitted with non-reversible hydraulic couplings (Fig. 22), whereas in the "Milwaukee" the engines are rigidly coupled.

### M.S. "Shinsu Maru" (ship speed 16 knots).

Two eight-cylinder engines, each developing 1,350 b.h.p. at 420 r.p.m., coupled through Vulcan couplings to reduction gearing, giving a propeller speed of 90 r.p.m.

### M.S. "Siantar" (8,438 tons gross).

The "Siantar" was originally fitted with geared turbines having a speed of 13 knots. After conversion to geared oil-engine drive the speed was increased to $15\frac{1}{2}$ knots. Two Schelde-Sulzer six-

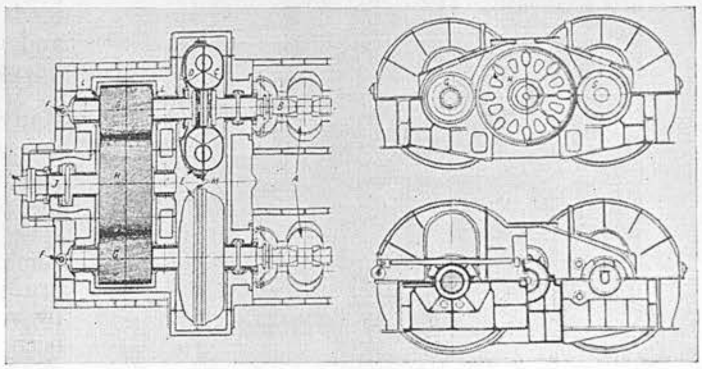

Fig. 22.—"St. Louis"—gears and transmission.

cylinder double-acting two-stroke engines developing 3,650 b.h.p. at 215 r.p.m., connected through Vulcan couplings to reduction gearing reducing the propeller speed to 98 r.p.m., are fitted.

MOTOR TUG "ZWARTE ZEE".

Two six-cylinder Werkspoor engines of the two-stroke single-acting type, developing 1,680 b.h.p. each at 275 r.p.m., are connected through Vulcan coupling and reduction gearing to the propeller, which has a speed of 120 r.p.m.

M.S's. "MANOERAN" AND "MADOERA" (14,000 TONS).

Originally designed for steam engines with a speed of 12 knots, the above ships have recently been converted to geared oil-engine drive, increasing the speed to 15 knots. Two Werkspoor-Sulzer single-acting two-stroke engines, each developing 3,350 b.h.p. at 225 r.p.m., are coupled to the propeller through Bibby couplings and gearing reducing the propeller speed to 86 r.p.m. The gears are of double helical Demag type, having an efficiency on test of 99·1 per cent. Fig. 23 shows the layout of the gears and Bibby couplings.

M.S. "MODJOKERTO" (8,396 TONS GROSS).

This ship of the Rotterdam Lloyd Steamship Co. was originally fitted with a geared turbine of 3,500 s.h.p. New machinery consisting of two 3,500 h.p. six-cylinder double-acting two-stroke M.A.N. engines 530×760mm. at 215 r.p.m., coupled

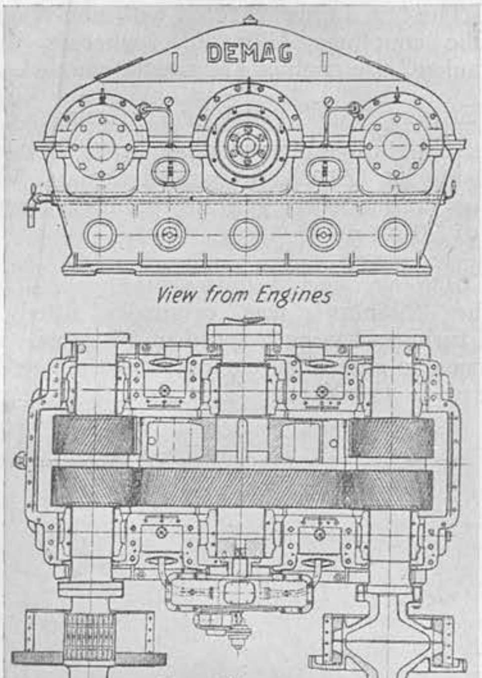


FIG. 23.—M.S. "Manderan". Showing gears and Bibby couplings.

through Vulcan couplings and geared to the same shaft was fitted, giving a propeller speed of 98 r.p.m. (i.e., reduction 2·3 : 1). Speed was increased from 11½ to 15½ knots.

"ADMIRAL SCHEER".

This is the second motor battleship of the Deutchland class, in which the machinery consists of eight M.A.N. nine-cylinder double-acting two-stroke engines, 420mm.×580mm., of welded frame construction. 7,100 b.h.p. is developed at 450 r.p.m., the engines being coupled in groups of four through Vulcan couplings and single reduction gearing to the two propeller shafts rotating at 350 r.p.m. The actual power at the propeller, allowing for gearing losses, is 54,000 b.h.p.

From the short list of recent geared installations given above, which is by no means exhaustive, it will be apparent that the geared oil-engine drive is not confined to any one type of engine or ship. Although several four-stroke single-acting engines have been geared to the propeller and experience extending over ten years has testified to the soundness and reliability of such installations, it will be readily agreed that on account of the small torque variations the two-stroke (and more especially the double-acting two-stroke) is eminently the more suitable type of oil engine for the purpose. With increased experience and confidence in the design and running of the double-acting two-stroke, there is no doubt that this type will take precedence over its competitors for this particular type of installation, since it satisfies economic considerations. The chief advantages of the geared drive apart from those of cost, space and headroom, lie in the possibility of grouping a number of relatively small engines by coupling to one or more shafts. By such grouping, it becomes possible to run a greater or lesser number of engines at the most economical rating according to variations in power requirement and, at the same time, the possibility of complete breakdown becomes more remote. Not least among the advantages of a system utilizing high-speed engines is the comparative ease of repair on account of the reduction in size and weight. Inspection is simplified and from the psychological point of view the ease of handling tends to ensure that inspection is made voluntarily and not as is often the case with the slow-speed heavy engine, only when necessity compels.

## Railway Transport.

Although petrol-driven shunting locomotives and railcars have found considerable application in the past, it is only since the advent of the high-speed airless-injection oil engine that the internal combustion engine has become a competitor to the firmly-established steam engine. The petrol engine, by virtue of its high speed and high b.m.e.p., has been the principal factor in the extensive adoption of the oil engine for railway service, since, in spite of the initial disadvantage of a relatively high

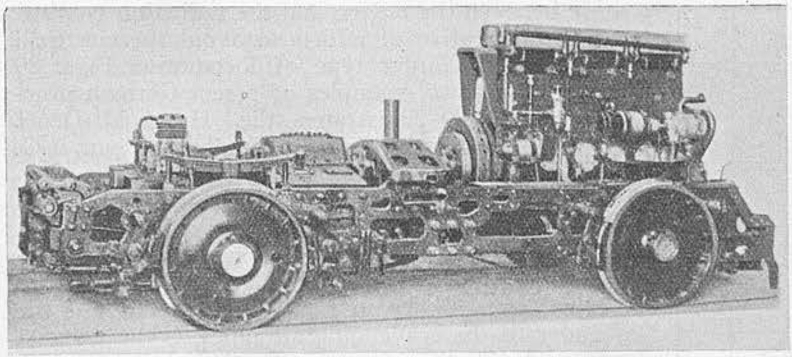

FIG. 24.—Driving bogie of French railcar.

frictional road resistance of about 70lb. per ton as compared with a rail resistance of about 3lb. per ton, it has enabled the development of speedy, comfortable, reliable and flexible road transport.

The railways are very much alive to the benefits of the oil engine as a means of speeding and cheapening rail transport. An endeavour will therefore be made to determine the place of the oil engine in present and future railroad development. As in every other case of transport, the type of engine to be used is determined by the factors of suitability for the particular type of service, acceleration, speed, comfort and economy. As far as the first and second are concerned, the oil engine is definitely unsuitable for railway traction because the demands of the service require high starting torque and ability to meet large and sudden variations in load. With its constant torque characteristic, the oil engine is unable to meet these exacting requirements and it becomes necessary to interpose some form of energy transformer between the engine and the driving wheels of the locomotive in order to provide the requisite flexibility. For the smaller types of railcar and shunting locomotives using engines up to 250 b.h.p. several types of flexible couplings are available and gear transmissions, hydraulic couplings, hydraulic torque converters and electrical transmissions have been used successfully on these types of locomotive.

A typical rail car that is used extensively on the French railways has seating capacity for 62 persons and loaded weighs 35 tons. The power unit consists of two Saurer engines, each developing 130 b.h.p. at 1,500 r.p.m., located in the bogies at each end. The drive is transmitted to the wheels through a gear-box giving four forward speeds and reverse. Gear changing is effected through an oil pressure system which can be controlled from either cab. The complete driving bogie of a similar type of railcar used on the Paris-Orleans Railway, developing 185 b.h.p. at 1,000 r.p.m., is shown in Fig. 24.

The introduction of the hydraulic coupling and the hydraulic torque converter opened up a new method of coupling the engine to the driving wheels, and systems incorporating these drives are being increasingly used. The hydraulic coupling used for rail traction is known as the traction type and differs somewhat from the marine type previously described. The driven member B (Fig. 25) has an annular reservoir C formed concentrically with the driven shaft D; fixed to the driven member and connecting the dead area E of the system to the reservoir are scoop tubes K. At low speeds of the driven member B, oil flows from the dead area E to the reservoir C, but as the speed increases the centrifugal force transfers the oil from the reservoir back into the coupling. These actions take place during a very small range of speed and enable the coupling to function directly with the engine speed and ensure a smooth take off. Fig. 26 indicates the slip of this type of coupling plotted against speed expressed as a percentage of the maximum.

A 130 h.p. oil-engine-driven railcar supplied to the Great Western Railway Co. by Messrs. Hardy Motors, Ltd., of Southall, uses a transmission system embodying the hydraulic coupling and the Wilson pre-selective gear-box. This car weighs 20 tons and is designed to seat 70 passengers. The engine is of standard A.E.C. type, delivering 130 b.h.p. at 2,000 r.p.m. and the drive to the axles is through the hydraulic coupling and Wilson epicyclic four-gear box to a reverse gear and from thence the final drive is taken to the axles by worm gearing. As both axles on the bogey are driven, an interaxle differential gear is provided in the leading axlebox in order to take up any irregularities in the drive due to any slight differences in driving wheel diameters.

Another type of flexible coupling which has already given encouraging performance is the hydraulic torque converter of the Lysholm-Smith type. This converter is designed on the turbine principle and consists of a pump wheel directly driven by the engine shaft and a turbine composed

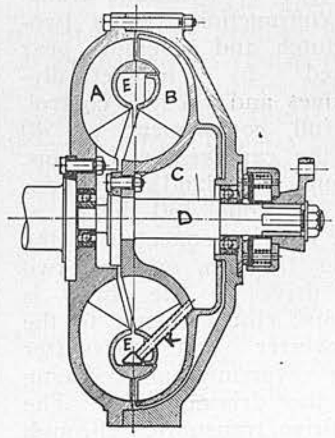


FIG. 25.—Hydraulic coupling (traction type).

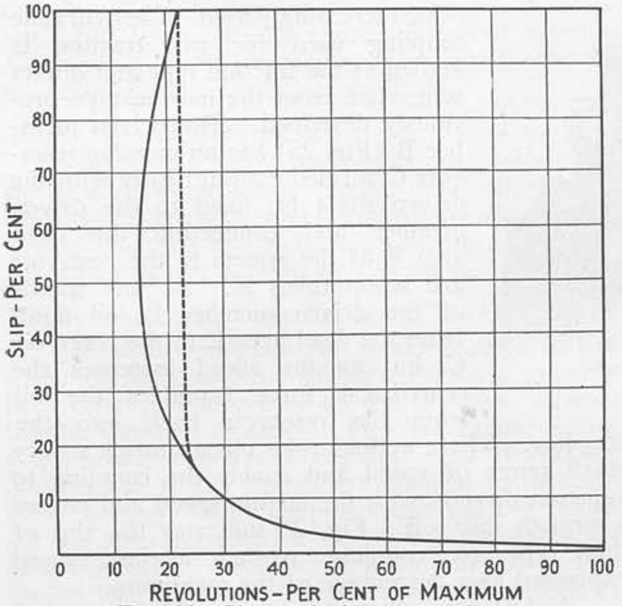

FIG. 26.—Slip of hydraulic coupling.

of three rings and connected to the driven shaft. Between the turbine rings are interposed two stationary guide rings. From the pump the liquid passes through the first turbine ring and is reversed in a guide ring, proceeding to turbine ring number two, again being reversed in the second guide blade ring and finally passing through the last turbine ring, from where it returns directly to the pump. In the cross section shown in Fig. 27 a reverse gear is shown, but in the case of its application to the railway locomotive this gear is usually contained in a separate housing and actuated by vacuum cylinders. As will be seen from Fig. 28, the efficiency curve is flat over a considerable range of speed and consequently the necessary cooling surface to be supplied for cooling the working fluid is small.

A number of light railcars built by Messrs. Leyland Motors, Ltd., for the L.M.S. Railway and recently placed in commission successfully utilize this latter type of hydraulic transmission in conjunction with a two-way disc clutch and reversing gear interconnected by magnetically-operated valves and vacuum control. With its full complement of 40 passengers the car weighs 13 tons. The power unit, a Leyland six-cylinder oil engine, develops 130 b.h.p. at 2,000 r.p.m. and is connected to the driving axle through one of two alternative drives. The first is through a disc clutch thence to the torque converter and thereafter through the reversing and reducing gearing to the driving axle. The alternative drive transmitted through the second disc clutch gives direct drive between the engine and the reduction gearing.

As indicative of efforts to avoid the electrical drive for the larger type of locomotive Figs. 29 and 30 show two examples of recent German practice. Fig. 29 illustrates the Humboldt-Deutz 1,000 h.p. locomotive built by Humboldt-Deutz A.G. The engine consists of three double-acting two-stroke cylinders 380mm.×600mm. running at 250 r.p.m., directly coupled to the driving wheels. For starting, the cylinders are supplied with compressed air. A more detailed view of the engines and cooler is shown in Fig. 30, in which A is the cylinder, B the scavenging air pipes, C the exhaust manifold, D the silencer, E the radiator with electrically-driven fan, F the piston cooling water pipes, and G the starting air valves.

The power/weight ratio of the oil-engined locomotive or railcar will, of course, depend on two main factors. Firstly, on what extent it is desired to speed up existing services, and secondly, the type of service. For example, in the case of certain suburban services where traffic density is high but not high enough to warrant a fully electrified system and where stations are separated by short distances, high power/weight ratios are necessary. In such cases, acceleration counts more than maximum speed and it necessitates large powers. For instance, in the Leyland railcar previously described above, the power/weight ratio is 10 to 1 and the acceleration 3·25ft. per second per second. The high power weight ratio enables this railcar to attain a speed of 20 m.p.h. in 11 seconds and 50 m.p.h. in 50 seconds. It is doubtful whether high ratios of the above order can be justified in any type of oil-engined railcar. Certainly the services in which it could be successfully and economically employed only exist as a thin dividing line between suburban and urban requirements. The average power/weight ratio of the present type of railcar employed both in England and abroad is about 5 b.h.p. per ton, producing an acceleration

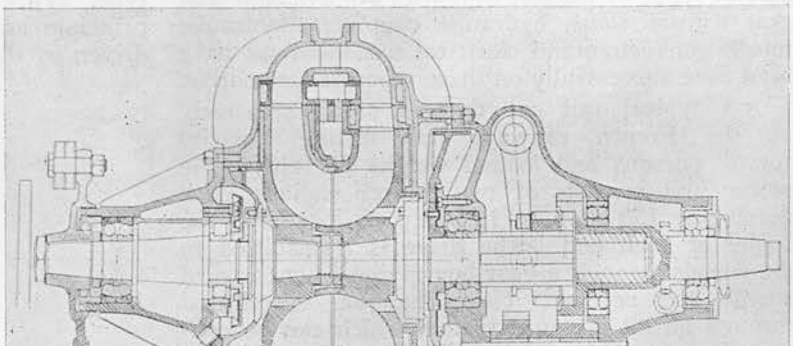
FIG. 27.—Lysholm-Smith torque converter.

of about 1·6ft. per second per second, which is just a little higher than that of the average motor bus.

While gear transmission with or without the interposition of the hydraulic coupling will doubtless find increased scope in future railcar development, there can be no question that electrical transmission represents the most logical means and the best technique in securing for the oil engine those steam engine characteristics so essential to satisfactory railroad performance. This is well established by the fact that out of the total number of oil-engined locomotives or railcars in service throughout the world over 80 per cent. employ electrical transmission.

Having now briefly traced the various methods available to ensure flexibility to the oil engine and thus render it applicable to railway purposes, it remains to consider the remaining factors which determine its suitability for widely varied types of service.

On the score of reliability, experience gained in Canada, South America and on the Continent over the last ten years, is sufficient to prove that on the whole a reliability factor of about 85 per cent. can be obtained with the oil-electric type, and a factor slightly less with the geared type; there is nevertheless a lack of consistency in operating results which indicates that too much care cannot be exercised in the selection of the design for any given type of service. Considering that the oil engine is in many cases supplanting steam locomotives whose designers and operators have the benefit of over a hundred years' experience upon which to draw, the results have been very satisfactory, and as experience is gained and types are standardized for the various services, there is no reason to doubt that the oil engine will rank equally with the steam engine as a consistently reliable prime mover for railway work.

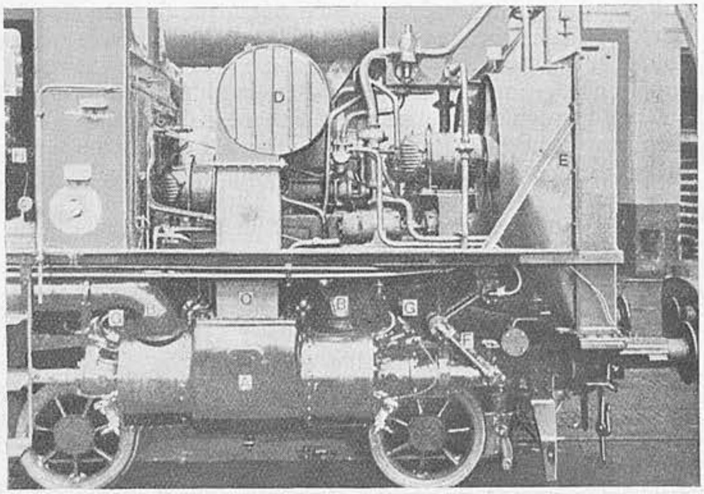

Fig. 30.—Humboldt-Deutz locomotive—details of power unit.

On grounds of economy much has been published for and against the oil-engine locomotive. Excluding maintenance, on which very little reliable data exists, the economy or otherwise of the oil-engined locomotive is governed by three main factors. Assessed in order of importance, these may be placed as follows:—

(*a*) The position of the country concerned with regard to mineral wealth.

(*b*) The relative cost and value of coal and oil.

(*c*) The relative cost of the oil engine as compared with the steam engine.

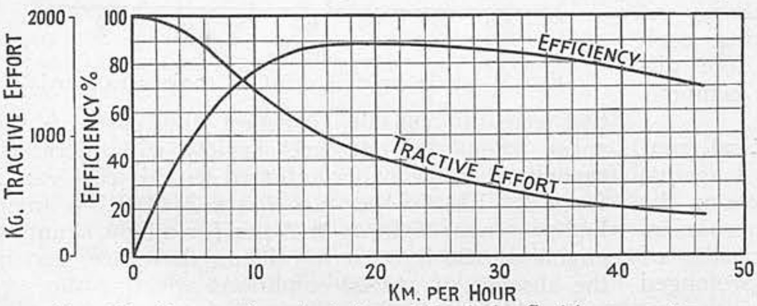


Fig. 28.—Power characteristics of Lysholm-Smith converter.

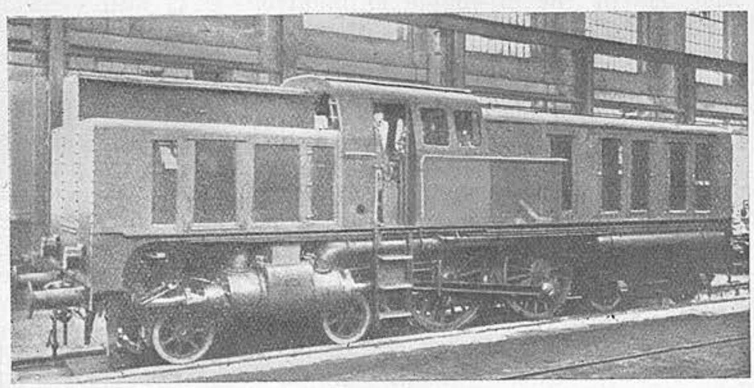

Fig. 29.—Humboldt-Deutz 1,000-h.p. locomotive.

Considering these influences in greater detail and in the order given above, it is pertinent that the greatest developments in oil engine railway transport have taken place in those countries having natural oil resources. In others, deficient in both coal and oil supplies and in certain countries which, although containing considerable coal resources, must necessarily, either on account of their poor quality, internal transport difficulties, or the distance of the coal fields from manufacturing centres, import coal, development has been considerable.

In Germany, where perhaps the most extensive development has recently taken place, the loss of coal deposits under the peace settlement and possibly the political policy of securing independence of internal transport facilities in the event of the Ruhr coalfields being lost in war, must be regarded as supplying the incentive. The oil-engined battleships of the "Deutchland" class are further evidence that Germany is pinning her faith to oil and the question of ensuring security of supplies must be carefully considered in her future policy. Considering the state of the world to-day and the unsettled conditions prevailing, the application of the oil engine to general rail transport in such fortunate countries as Great Britain, possessing plentiful resources of good coal, can only be looked upon as a gesture or an experiment. Desirable as it may be from the point of view of economy, it is inconceivable that any Government will permit of anything like a general changeover from the existing steam-engined transport to oil transport. The most that can be expected in such countries, apart from considerations of economy, is an increase in the number of shunting engines and railcars operating on branch and suburban lines and in certain cases a supplementary service to main line engines. It is possible, of course, that hydrogenation of coal may become an economic possibility, but present methods would indicate that, due to remissions of Government tax on petrol, hydrogenated petrol may compete with the imported fuel.

From a purely defensive view point, therefore, it would appear that for countries such as Great Britain, possessing widely distributed good-quality coal supplies and not possessing oil supplies, railway transport must be mainly supplied by steam traction or alternatively by complete electrification. On the other hand, in those countries possessing oil supplies or in those countries importing coal, present indications tend to show that the oil-engined locomotive will become a serious competitor of steam.

Apart from political and geographical considerations, the factor which militates most against a more universal adoption of the oil engine on the railway is its high initial cost. Until the mechanical or hydraulic gear has proved that it is capable of efficient and economical service over prolonged operating periods, the alternative is electric transmission. Electric transmission is costly and as standard traction motors are used it is unlikely that any great reduction in the price of the electrical equipment can be expected. In a lecture* delivered before the Institution of Mechanical Engineers, Lt.-Col. F. R. Fell considered that with reasonable outputs the oil-electric locomotive using a high-speed oil engine could be manufactured at a cost of £13 10s. 0d. per b.h.p. Of this amount, the oil engine accounted for £2 10s. 0d., the installation including framing, cooling, etc., £3, and the electrical equipment £8. In the subsequent discussion, doubt was expressed concerning the possibility of reaching such a low figure of £13 10s. 0d. per b.h.p., but recent production costs would indicate that the sum is not very wide of the mark.

Taking for purposes of comparison a figure of £13 per b.h.p., it becomes possible, after taking into account the ruling prices of oil and coal and the relative efficiencies of the oil and steam engines, to determine the possible extent to which the oil-electric locomotive may replace the steam locomotive. In this connection, it has been often pointed out that the oil locomotive is a twenty-four hours per day, six days per week unit and can therefore often replace two or more steam locomotives. While such is undoubtedly the case, the services in which such advantages could be utilized to the full are not numerous. One of the most important factors determining the economics of the question is the fact that while the cost of the oil locomotive remains sensibly constant the cost of the steam locomotive decreases rapidly with increased capacity. In the paper referred to, this cost varies from £4 10s. 0d. per h.p. for a 2,000 b.h.p. main line passenger train, to £14 per for a light shunting engine of 250 h.p. It is evident, therefore, that in the absence of special conditions affecting the service, there is a definite limit to the useful adoption of the oil-engine locomotive. It is also evident that for oil-engined shunting locomotives and railcars up to 500 b.h.p. there is ample scope and it is in this direction that the greatest development has taken place.

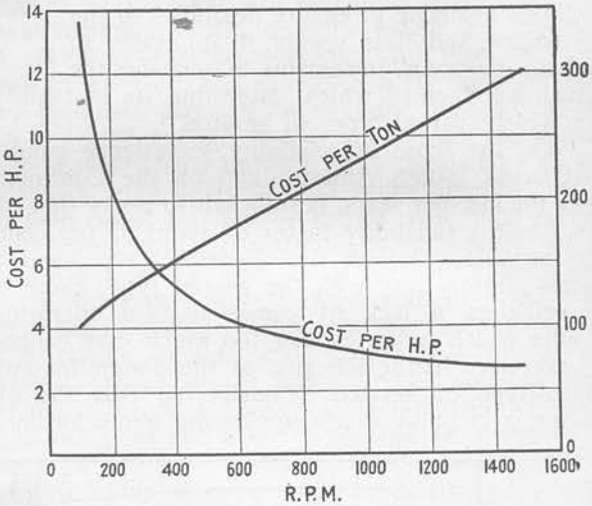


FIG. 31.—Relation of cost of oil engines to speed.

The employment of higher engine speeds serves both to cheapen and lighten the engine, and present indications would lead to the conclusion that oil engines suitable for locomotive and railcar service running at speeds between 1,000 and 1,500 r.p.m. are likely to become standardized. The decrease in cost with increase in speed is indicated by Fig. 31 taken from Lt.-Col. Fell's paper. A further saving in cost and weight could be accomplished by supercharging. In the four-stroke marine oil engine, supercharging is now standard, and with an

*"Compression-Ignition Engine and British Railways". Proc. Inst. Mech. Eng., Jan., 1933, p. 3.

electrically-driven centrifugal supercharger a boost of some 30 per cent. could be obtained for starting and grade work, thus enabling a substantial reduction in the weight and cost of the engine.

## Aircraft.

The chief requirements of a successful engine for aircraft propulsion are lightness combined with maximum power output. It will be evident from previous consideration, that in comparison with the petrol engine, the oil engine having a high ratio of maximum pressure to b.m.e.p. suffers considerable disadvantage from the point of view of a high power/weight ratio. This disparity has become more pronounced in recent years in that with the introduction of volatile fuels of high octane rating, higher useful compression ratios can now be used in the petrol engine, resulting in increased efficiency, economy and output.

There are, however, so many advantages inherent in oil-engine design and operation, that considerable attention has of late years been devoted to its development for aircraft propulsion. Immunity from fire risk, simplicity of design, maintained output at high altitudes, low working temperatures and high efficiency combined with the present low cost of oil fuel being the chief characteristics which have given the incentive to this development.

The Junker and Maybach engines in Germany and the Packard in America were among the first successfully developed aeroplane oil engines, and considerable experimental work has been carried out by British engineers in conjunction with the Air Ministry, notably on the Beardmore, Rolls-Royce "Condor" and Bristol "Phoenix" engines.

A Rolls-Royce twelve-cylinder "Condor" engine has been converted to use fuel oil. The cylinders, which are individually mounted, have strengthened flanges and two Bosch type injection pumps are fitted. As a petrol engine, the weight was 1,395lb., or approximately 2·9lb. per h.p. After conversion, when developing 480 b.h.p. at 1,900 r.p.m., the weight was 1,513lb., or 3·15lb. per h.p. The fuel consumption was 0·423lb. per b.h.p./hour.

The Bristol "Phoenix" is of the radial air-cooled type, and develops 470 b.h.p. at 1,900 r.p.m. The complete weight is 1,090lb., or about 2·63lb. per b.h.p.

During a test flight carried out on May 14th, 1934, a "Wapiti" general purpose aeroplane weighing 4,564lb. complete with pilot and full equipment attained a height of 27,460ft., creating a world's record for an oil-engined aeroplane. From comparative test records, it was shown that power at altitude was maintained (as one would naturally expect) much better than is the case with the petrol engine.

Leading particulars of the "Phoenix" engine as fitted to the Westland Wapiti on its record-making flight are given in Table IV,* and Table V* shows a comparative tabulation of power, consumption and weight for compression-ignition and petrol nine-cylinder air-cooled radial engines. From this

TABLE IV.

*"Bristol" Phoenix Engine.*

| | |
|---|---|
| Type ... ... ... ... | 9 cyl. air-cooled |
| Bore and stroke ... ... | 5·75in.×7·5in. |
| Swept volume ... ... | 1,753 cub. in. |
| Engine r.p.m. ... ... | 1,900 normal 2,000 maximum |
| Airscrew ratio ... ... | 0·655 engine speed |
| Rated power at normal r.p.m. ... ... ... | 415 b.h.p. |
| Take off power at normal r.p.m. ... ... ... | 470 b.h.p. |
| Power at maximum r.p.m. | 430 b.h.p. |
| Weight complete ... ... | 1,090lb. |
| Fuel used ... ... ... | Persian gas oil (specific gravity 0·839) |

TABLE V.

*Compression-Ignition and Petrol Nine-cylinder Air-cooled Radial Engines.*
*Comparative Tabulation of Power, Consumption and Weight Data.*

| Type of engine | Relative maximum output per litre. | | Cruising fuel consumption. | Weight per cruising b.h.p. Engine gross to B.S. Specn. No. 185 plus fuel and oil. | | | | |
|---|---|---|---|---|---|---|---|---|
| | Take off. | Flight. | lb./b.h.p. hr. | 2 hrs. | 4 hrs. | 6 hrs. | 8 hrs. | 10 hrs. |
| Compression-Ignition, 1934 ... | 100 | 100 | 0·390 | 4·16 | 5·05 | 5·95 | 6·84 | 7·73 |
| Petrol, 1930 ... ... ... | 126 | 140 | 0·543 | 4·31 | 5·56 | 6·80 | 8·05 | 9·30 |
| Petrol, 1934 ... ... ... | 163 | 174 | 0·491 | 3·88 | 4·99 | 6·10 | 7·21 | 8·32 |

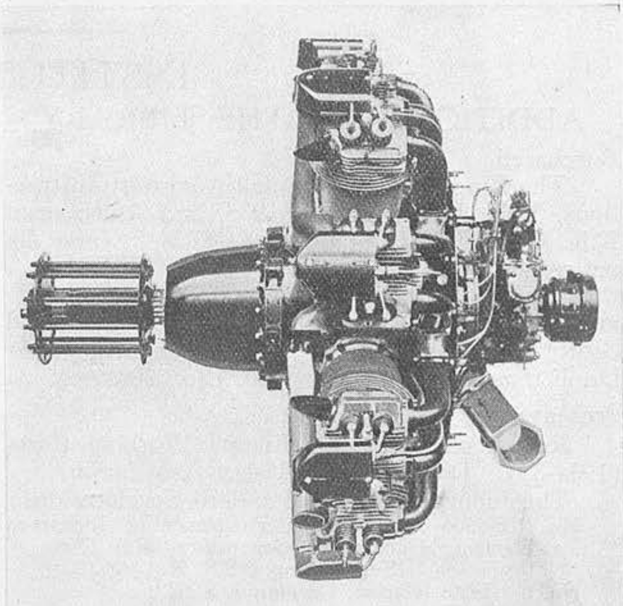

FIG. 32.—Bristol Phoenix air-cooled oil engine.

*Through courtesy Messrs. Bristol Aeroplanes, Ltd.

tabulation, it will be evident that the compression-ignition-engined aeroplane will show to advantage in all-up weight on a flight of a little over four hours' duration.

Encouraging as these figures may be, it must be realized that oil-engine design as applied to aircraft is only in the experimental stage, and it is quite conceivable that when full advantage is taken of the lower working cylinder temperature obtaining in this type of engine, and the two-stroke air-cooled radial engine developed as a consequence, it may well supersede the petrol engine in the race for a high power/weight ratio.

Conclusion.

Oil-engine development has been a continuous process since the eighties of last century. In any historical review of the subject, the names of Akroyd Stuart and Rudolph Diesel must stand out pre-eminently, but there are many workers of the present generation whose names will doubtless stand out just as clearly in the years to come when the history of the oil engine is written for our children to read.

The oil engine to-day is making rapid inroads upon the fields of power production and transport. Its success for sea-going vessels is assured. For road transport, it has been proved the successor of the petrol engine, provided the present supply of cheap fuel can be maintained.

Railway transport demands much from the power unit. It remains to be shown that the oil-engined locomotive can supplant or even compete with the steam locomotive for general rail transport.

In the air, the oil engine suffers from comparatively low power/weight ratio. In the near future, however, it will doubtless be used to an increasing extent for heavy commercial and long-distance passenger aircraft, and when developed as an air-cooled two-stroke engine its adoption may become universal.

# INSTITUTE NOTES.

## ADDITIONS TO THE LIBRARY.

Purchased.

The King's Regulations and Admiralty Instructions, Vol. II, Appendices, 1935, and Amendment K.R. 6/36. H.M. Stationery Office, 2s. and 4d. net respectively.

"Airless Injection Heavy-oil Engines". Revised translation by James Calderwood, M.Sc., and G. R. Hutchinson, from the German original of Dr. F. Sass. Chapman & Hall, Ltd., 30s. net.

Presented by the Publishers.

Report on Heavy-oil Engine Working Costs (1934-35). Diesel Engine Users Association.

The following British Standard Specifications:

No. 168-1936. Electrical Performance of Industrial Electric Motors and Generators with Class A Insulation. (Superseding B.S.S. No. 168-1926).

No. 681-1936. Carbon Chromium Steel.

No. 682-1936. 3 per cent. Nickel Chromium Case-Hardening Steel.

No. 693-1936. Oxy-Acetylene Welding as Applied to Steel Structures.

No. 689-1936. Railway Mechanical Signalling Apparatus.

Transactions of The Institution of Mechanical Engineers, Vol. 131, containing the following:—

"The Strength of Metals under Combined Alternating Stresses", by Gough and Pollard.

"Technical Peculiarities of the Liner 'Normandie'," by Marie.

"The Utilization of Creep Test Data in Engineering Design", by Bailey.

"Muscles and Nerves: The Maintenance of Posture, the Development of Power and the Transmission of Messages in the Body", by Hill.

"Damping Influences in Torsional Oscillation", by Shannon.

"Stress Waves in the Tyres of Locomotives", by Coker and Salvadori.

"Hysteretic Influences on the Whirling of Rotors", by Robertson.

"The Embrittlement of Alloy Steels", by Lea and Arnold.

"Bulk Handling of Petroleum", by Graham.

"Mechanical Equipment of Modern Gas Works", by Johns.

"On Keeping Abreast of Technical Development", by Cave-Brown-Cave.

"A Survey of Engineering and Metallurgical Progress", by Wolstenholme.

"Heat for Advance Students", by the late Edwin Edser, A.R.C.Sc. Revised edition by N. M. Bligh, A.R.C.Sc., A.I.C. Macmillan & Co., Ltd., 487pp., illus., 6s. net.

This is a new and thoroughly revised edition of a book which has deservedly ranked as a standard for many years. While the essential character of the book has been preserved, the treatment of the subject, both theoretical and experimental, has been brought into line with modern knowledge and requirements; for example, in place of the somewhat primitive cardboard tube and lead shot method of determining Joule's equivalent we are now given a well illustrated discussion of Callendar's method and apparatus. In accordance with normal lecture-room practice to-day, elementary calculus is used freely in the treatment of such matters as work performed in compressing a gas, adiabatic transformations, the Carnot cycle, and entropy. New aspects of the subject have been introduced, and the final chapter, on radiation, leads up to the fringe of the quantum theory.

The engineering student who familiarised himself with the main contents of this book, particularly those parts relating to thermodynamics, would rest his future studies of the theory of heat engines upon a very secure foundation.

"Engineering Questions and Answers", Volume I. Emmott & Co., Ltd., 176pp., illus., 6s. net.

This is a book which should not be hastily judged by its title, for "Engineering Questions and Answers" does not introduce the reader to that doleful type of textbook, now fortunately rarely encountered, which attempts to instruct the student by asking him innumerable questions and supplying forthwith the innumerable answers, all equally terse and often incomprehensible.